Elements of

APPLIED ENERGY

by

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SECOND PRINTING

D. VAN NOSTRAND COMPANY, INC.

TORONTO NEW YORK LONDON

NEW YORK
D. Van Nostrand Company, Inc., 250 Fourth Avenue, New York 3

TORONTO
D. Van Nostrand Company (Canada), Ltd., 228 Bloor Street, Toronto 8

LONDON
Macmillan & Company, Ltd., St. Martin's Street, London, W.C. 2

First Published April 1947

Reprinted March 1948

PRINTED IN THE UNITED STATES OF AMERICA

Preface

Energy is a most important and fundamental entity. It is observable in the forms of electricity, heat, radiation, work. *Elements of Applied Energy* intends to acquaint the reader with the nature and interrelation of these forms, then show him the how and why of machines and other equipment in which energy plays a part.

This book is the outgrowth of some years of frustration—when, repeatedly, the author sought, with indifferent success, to implant a sound knowledge of Thermodynamics and of Heat Power in the minds of many an embryo engineer, beginning usually at the second-year level. Unfortunately, the abstractions and the mathematics of thermodynamic science were usually sufficient to vanquish a goodly percentage of each class. Finally, after deciding that a general survey of applied energy, its equipment, and its machines, was more appropriate to sophomore status, the author could locate no text suitable to the scope he envisaged, and so undertook to prepare this volume. It is hoped that others may have arrived at similar conclusions, and that the book will serve them, also.

A thoroughgoing thermodynamics course seems now, to the author, to be inappropriate for those who pursue lines of study only casually involved with heat engines. His viewpoint is that, for those to whom a single course is not only the introduction to, but also the termination of, formal heat power study, the greatest benefit is secured from a less scientific but more descriptive survey of energy applied to the uses of man. It is also hoped that, with the orientation this engenders, the potential heat power specialist may thereafter approach a formal study of Thermodynamics more purposefully and with less unreasoning apprehension. So this volume was prepared with the thought of its becoming a useful terminal study in applied energy for some, and a suitable introduction to heat power for others. It is not a Thermodynamics text, nor is it exclusively a Heat Power text. Since the author has endeavored to avoid over-emphasis in any direction, it is hoped that a balanced presentation has been accorded the subject. Chapters 1–5 are necessary background; Chapters 6–7 cover energy applied to other purposes than producing work; Chapters 8–11 cover work from direct combustion; and Chapters 12–15 work from indirect combustion.

It is presumed that the reader has no more mathematical background than that afforded in a good command of algebra, plus an elementary acquaintance

with trigonometry. No prior knowledge of the calculus is assumed, and none is developed herein.

The following suggest different degrees of utilization of the text content. The author intended to create a textbook for a standard sixteen-week course, a chapter being provided for each week's study. Some may elect to modify this by the omission of some sections from each chapter. For an average sixteen-week course it may be desirable to omit one or more of Chapters 7, 10, 11, or 15. Where thirty-two weeks are available for its study, the full text can be covered in a leisurely fashion.

The text affords ample practice work for the student—work that may be initiated at the outset of the course of study and continued regularly throughout. The nature of the problems will be found to be both mathematical and graphical. An effort was made to put the problems in balance, i.e., to compose them so that their individual solutions will require approximately the same effort and time. They are not particularly difficult, but each is designed to aid in developing some facet of the ability to visualize and analyze.

To the several firms who have so cooperatively furnished illustrative material, the author expresses again his indebtedness. The D. Van Nostrand Company has kindly consented to the use of many illustrations the author prepared for its *Scientific Encyclopedia*. Finally, I acknowledge with especial pleasure the invaluable assistance of my wife, Mary Genevieve, both in encouragement to carry the work to completion, in the preparation of the manuscript, and while the book was in press.

F. T. M.

Charlottesville, Virginia
January, 1947

Contents

Symbols

A	Area. Atomic mass number.
a	Acceleration.
C	A constant. Percent clearance. Centigrade.
c	Specific heat. Velocity of light.
D	Diameter. Drag.
d	Distance. Density.
E	Voltage. Exhaust lap.
e	Energy, general.
F	Force. Fahrenheit.
f	Power factor.
g	Acceleration of gravity.
H	Velocity head. Height.
$\mathscr{H}$	Total fluid head, feet.
H.V.	Heating value.
h	Enthalpy. Heat content. Altitude.
I	Amperes.
J	Mechanical equivalent of heat.
K	Coefficient of conductivity. A constant.
L	Distance. Stroke.
L.H.V.	Lower heating value.
M	Mols. Rocket mass.
m	Mass.
N	Rotative speed, revolutions per unit time.
n	Polytropic gas exponent. Number of an item.
P	Unit pressure, lbs. per sq. ft. Power.
$\mathscr{P}$	Total fluid head, lbs. per sq. ft.
p	Unit pressure, lbs. per sq. in.
Q	Transferred heat energy.
R	Gas constant. Rankine. Cut-off ratio. Resistance.
r	Compression ratio. Expansion ratio. Radius.
S	Steam lap.
s	Entropy.
T	Absolute temperature.
t	Fahrenheit or Centigrade temperature. Time.
U	Thermal conductance.

u	Internal heat energy. Tangential velocity. Rocket velocity.
V	Volume.
v	Velocity. Specific volume in vapor tables.
W	Work energy.
w	Weight. Velocity component, tangential.
x	An unknown quantity. Vapor quality.
y	Manometer reading. Height above datum.
Z	Atomic number. Combustion pressure ratio.
z	Gas coefficient.
Δ (Delta)	An increment.
Σ (Sigma)	An operator, meaning "to sum up."
ω (Omega)	Rotative speed, radians per unit time.
γ (Gamma)	Adiabatic gas exponent.
η (Eta)	Efficiency.
ρ (Rho)	Mass density.

CHAPTER 1

Introduction to Applied Energy

Today, in the civilized regions of the world, mankind uses mechanical energy liberally to supplement, replace, or extend its own limited physical powers and those of the draft animals which were first domesticated in previous historical eras. This mechanical energy is mainly derived from *heat*, which is another form of energy, easily generated by the combustion of a fuel. Often the mechanical energy is transformed into electrical energy. This book is a survey of the nature of energy, of transfers and transformations of it, and of its production for the multitudinous uses of our mechanized civilization. The practical treatment rests upon and extends a phase of physical science, to wit: the phenomena and laws of *energy*. This field of science, known as *thermodynamics*, is often divided into pure and applied thermodynamics. The former can become highly complex, mathematically and otherwise; the latter may be technically abbreviated to suit the circumstances. In this survey of applied thermodynamics, emphasis is put upon the form and operation of equipment rather than the thermodynamic science involved. It is believed that those who would proceed further into a study of thermodynamics will the more easily accomplish their ambition if they possess some understanding of the physical equipment in which the thermodynamic processes occur. Furthermore, for many readers to whom the technology of heat power plants is not a primary field of study or employment, this book can be the path to cognizance of a great and vital phase of modern technology without needless entanglement in a maze of theories, postulates, laws, and mathematical derivatives upon which construction and performance are founded.

1-1. Physical Measurement. Physics and engineering are quantitative sciences which usually deal with things that are measurable in some physical units. Examples of these quantities are length, mass, pressure, volume, temperature, and time. There are many of these physical quantities. Those of greatest importance in the field of heat power will be described in some detail in subsequent sections. Frequently these quantities are properties of physical matter. They have a practical significance when they can be measured, either directly or indirectly. Length may be measured directly by use of a yardstick, but temperature must be gaged by its effects—as by thermal expansion of mercury in the bore of a thermometer. For most measurements, the pri-

mary quantities are length, mass, and time. Various combinations of these produce numerous secondary quantities such as volume, which is $(\text{length})^3$, and velocity, which is length/time. The two systems of mensuration in common use differ in the scale of the primary quantities of length and mass. The c.g.s. system is based on the centimeter and the gram, whereas the "English" system employs the foot and the pound. Pure science generally employs the c.g.s. system, engineering and industry the English system. The viewpoint adopted for this book dictates adherence to the English system of units.

1–2. Structure of Matter.* It is exciting to discover how all kinds of matter are ultimately found to be fashioned from the same few kinds of "building block." Just as a few basic kinds of brick, if in sufficient number, may be worked into many different buildings of varied architecture, so are all the different chemical elements, from aluminum to zirconium, built up of *atoms* made of just three kinds of "block." These are (1) the *electron*, (2) the *proton*, and (3) the *neutron*. Energy is the mortar which ties these blocks together. We cannot hope to see with human eyes the ultimate nature of the Universe, but that nature stands partially revealed today through physical experimentation of astonishing ingenuity, backed by the sheer intellect of the greatest reasoning age of history. These building blocks of nature (for all that is presently known) are the ultimate subdivisions of matter. Whether they are discrete particles or are wave-like in form is unimportant for the present purpose but, since the reader will visualize the necessary concepts of matter more easily if the electrons, protons, and neutrons are considered as definite particles, that viewpoint is recommended. The figures on page 3 follow this premise.

Though no one has seen an atom directly, or any of its parts, its physical dimensions have been thoroughly surveyed by indirect means. For example, it has been determined that the radius of an electron is 7.87×10^{-14} in. We now believe that atoms of different elements are simply varied combinations of the three kinds of particles. So an atom of carbon differs from an atom of iron only in the number and arrangement of its electrons, protons, and neutrons. In size, neutrons and protons have about the same mass,† but the electrons in comparison have practically no mass. The electron is a negatively charged (electrostatic) particle carrying 1.6×10^{-19} coulombs, and each proton carries an equal but positive charge. As its name implies, the neutron is electrically neutral.

According to our present views, every atom consists of a small heavy *nucleus* surrounded by a large empty region in which *electrons* move (in

* Portions of this section are taken from *Atomic Energy*, by H. D. Smyth.

† These masses have been measured, but the mass of the oxygen atom was chosen as the standard. This mass is 16. On the same scale a proton has a mass of 1.00758, a neutron 1.00893. The weight of a proton is 3.69×10^{-27} lb.

grouped orbits) somewhat like planets about the sun. This nucleus is composed of protons and neutrons and contains just about all the atom's mass. The number of electrons circulating about the nucleus equals the number

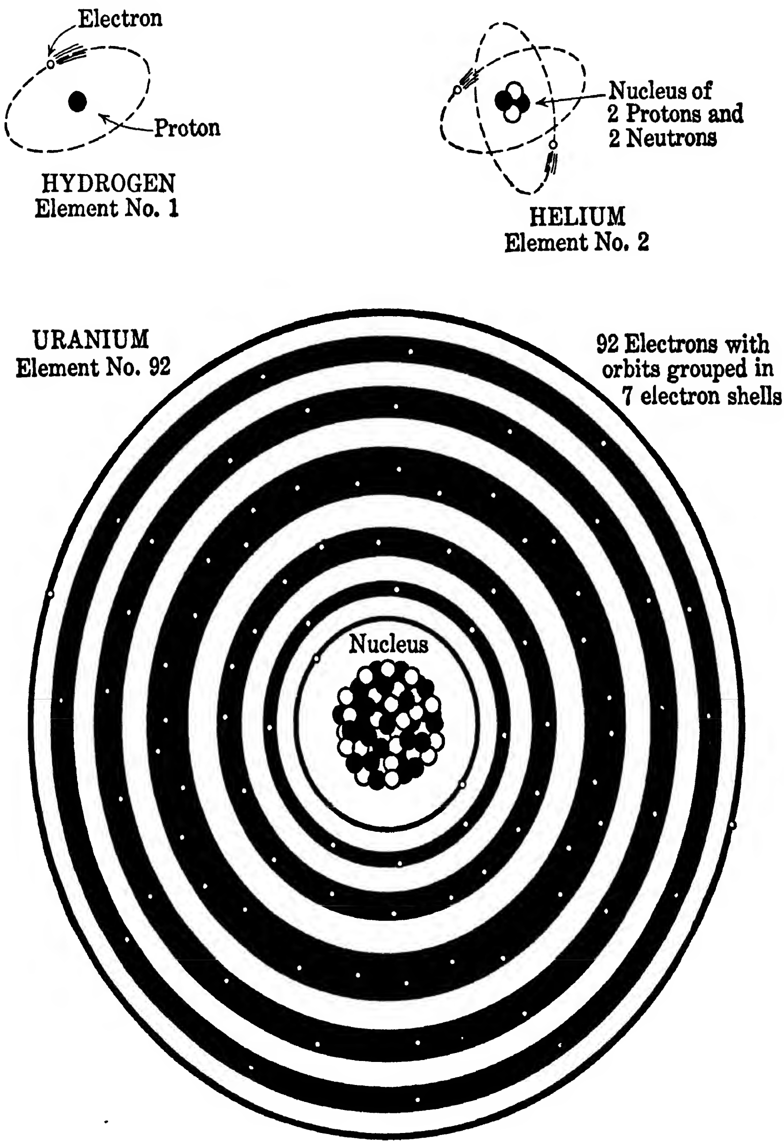


Fig. 1–1. Pictorial concept of atoms.

of protons in it so the atom, as a whole, has a net electric charge of zero. The number of neutrons approximates the protons in the lighter elements, but considerably exceeds them in the heavier.

The number of positive charges in the nucleus is called the atomic num-

ber, Z. It determines the number of electrons in the extranuclear structure, and this in turn determines the chemical properties of the atom. Thus all the atoms of a given chemical element have the same atomic number, and conversely all atoms having the same atomic number are atoms of the same element regardless of possible differences in their nuclear structure. The extranuclear electrons in an atom arrange themselves in successive shells according to well-established laws. Optical spectra arise from disturbances in the outer parts of this electron structure; X-rays arise from disturbances of the electrons close to the nucleus. The chemical properties of an atom depend on the outermost electrons, and the formation of chemical compounds is accompanied by minor rearrangements of these electronic structures. Consequently, when energy is obtained by oxidation, combustion, explosion, or other chemical processes, it is obtained at the expense of these structures so that the arrangement of the electrons in the products of the process must be one of lowered energy content. (Presumably the total mass of these products is correspondingly lower but not detectably so.) The atomic nuclei are not affected by any chemical process.

Not only is the positive charge on a nucleus always an integral number of electronic charges, but the mass of the nucleus is always *approximately* a whole number times a fundamental unit of mass which is almost the mass of a proton, the nucleus of a hydrogen atom. This whole number is called the mass number, A, and is always at least twice as great as the atomic number except in the cases of hydrogen and a rare isotope of helium. Since the mass of a proton is about 1800 times that of an electron, the mass of the nucleus is very nearly the whole mass of the atom.

Two species of atoms having the same atomic number but different mass numbers are called *isotopes*. They are chemically identical, being merely two species of the same chemical element.

The number of protons is equal to the atomic number, Z. The number of neutrons, N, is equal to the difference between the mass number and the atomic number, or $A - Z$. There are two sets of forces acting on these particles, ordinary electric coulomb forces of repulsion between the positive charges and very short-range forces between all the particles. Since these last forces are only partly understood, further explanation is not in order. Suffice it to say that combined effects of these attractive and repulsive forces are such that only certain combinations of neutrons and protons are stable. If the neutrons and protons are few in number, stability occurs when their numbers are about equal. For larger nuclei, the proportion of neutrons required for stability is greater. Finally, at the end of the periodic table, where the number of protons is over 90 and the number of neutrons nearly 150, there are no completely stable nuclei. These elements are characterized by *radioactivity*—which is atomic decay. As a sample of the orderly composition

of atoms examine the following data which represent the beginning and end of the periodic table of elements.*

Atomic Number, Z	Normal Element	Atomic Weight, A	Number of Protons	Number of Neutrons
1	Hydrogen, H	1	1	0
2	Helium, He	4	2	2
3	Lithium, Li	6.94	3	3, 4
4	Beryllium, Be	9	4	5
5	Boron, B	10.8	5	5, 6
	All intermediate elements are known.			
90	Thorium, Th	232	90	142
91	Protactinium, Pa	231	91	140
92	Uranium, U	238	92	146

Doubtless the reader discovers that the number of neutrons exhibited above is not always $A - Z$. This is because stable isotopes exist for some elements; the atomic weight is given for a mixture consisting mainly of the normal element, but with small quantities of the isotope present. For the pure normal element, $A - Z$ always equals the number of neutrons.

The one characteristic of neutrons which differentiates them from other subatomic particles is the fact that they are uncharged. This property of neutrons delayed their discovery, makes them very penetrating, makes it impossible to observe them directly, and makes them very important as agents in nuclear change. To be sure, an atom in its normal state is also uncharged, but it is ten thousand times larger than a neutron and consists of a complex system of negatively charged electrons widely spaced around a positively charged nucleus. Charged particles (such as protons, electrons, or alpha particles) and electromagnetic radiations (such as gamma rays) lose energy in passing through matter. They exert electric forces which ionize atoms of the material through which they pass. (It is such ionization processes that make the air electrically conducting in the path of electric sparks and lightning flashes.) The energy taken up in ionization equals the energy lost by the charged particle, which slows down, or by the gamma ray, which is absorbed. The neutron, however, is unaffected by such forces; it is affected only by a very short-range force, i.e., a force that comes into play when the neutron comes very close indeed to an atomic nucleus. This is the kind of

* On earth all elements except Nos. 1, 6, 7, 8, 11, 12, 13, 14, 15, 16, 17, 19, 20, 26 may be considered rare, some exceedingly so.

force that holds a nucleus together in spite of the mutual repulsion of the positive charges in it. Consequently a free neutron goes on its way unchecked until it makes a "head-on" collision with an atomic nucleus. Since nuclei are very small, such collisions occur but rarely, and the neutron travels a long way before colliding. In the case of a collision of the "elastic" type, the ordinary laws of momentum apply as they do in the elastic collision of billiard balls. If the nucleus that is struck is heavy, it acquires relatively little speed, but if it is a proton, which is approximately equal in mass to the neutron, it is projected forward with a large fraction of the original speed of the neutron, which is itself correspondingly slowed. Secondary projectiles resulting from these collisions may be detected, for they are charged and produce ionization. The uncharged nature of the neutron makes it not only difficult to detect but difficult to control. Charged particles can be accelerated, decelerated, or deflected by electric or magnetic fields which have no effect on neutrons. Furthermore, free neutrons can be obtained only from nuclear disintegrations; there is no natural supply. The only means we have of controlling free neutrons is to put nuclei in their way so that they will be slowed and deflected or absorbed by collisions. These effects are of the greatest practical importance in the release of subatomic energy.

So much for atoms. With rearrangements of the electrons, atoms may be joined to create *molecules*. In the process the atoms involved may gain, lose, or share electrons, but the nucleus is unaffected. In case the combination occurs through exchange of electrons, the formation of a molecule is something like this. The atom which has lost an electron becomes a positively charged *ion*, while that which gained it becomes a negatively charged one. Thus the molecule is held together by electrostatic forces of attraction between the ions. The ninety-two different atoms may be combined in an enormous variety of molecules, but not every combination is possible. For example, the gas argon is so indifferent that it refuses to enter into any combination whatsoever. It is chemically inert. But two oxygen atoms naturally join in the formation of the oxygen molecule, one of oxygen and two of hydrogen in the water molecule, and so on. This atomic composition of the molecule is described in the customary chemical symbolism for compounds, i.e., O_2 for oxygen, H_2O for water.

The molecules are of exceedingly small size since the atoms are small. It is said that there are 1.9×10^{25} molecules in a pint of water. In spite of the large numbers the molecules are not tightly packed together. Instead, considerable space separates adjacent molecules—space in which they may move about if they possess motion. That they do possess motion has been amply demonstrated, and this principle underlies the modern theory of heat energy. This will be developed later under the title "Kinetic Theory of Heat." The availability of intermolecular space is true of solids and liquids,

and especially so of gases (unless very highly compressed). Indeed, the relation of the size that the molecules of a gas bear to the space between them is not unlike that of the stars and planets to the firmament.

To understand most heat power processes one need not consider any smaller division of matter than the molecule, for ordinary thermal effects

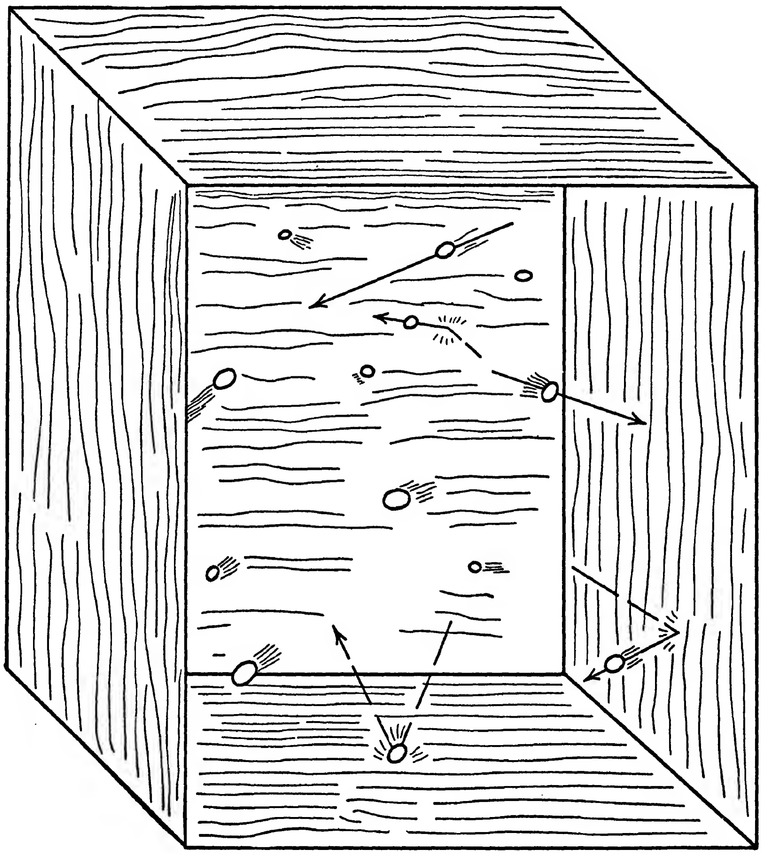


FIG. 1-2. Gas molecules in box with transparent end (molecular size exaggerated). Energy is possessed at the molecular level (contrasted to the atomic level) both in translation and in spin or vibration of the molecule.

can generally be explained by molecular activity. But *radiation* and the exploitation of *atomic energy* as a source of power are subjects for which some knowledge of atomic structure is a necessary foundation.

1-3. Temperature. The kinetic theory of matter having assumed molecular motion implies also a certain level of molecular kinetic energy. Temperature is the average molecular kinetic energy of translation and is measurable by various effects such as expansion, electrical resistance, or vapor pressure. Also the human sensory system is capable of perceiving temperature in a comparative way.

Temperature is one of the common words of English usage, but also one of the least understood. It is as likely to be incorrectly used by the average person as it is to receive accurate usage. While many temperatures are estimated with sufficient accuracy by simple observance of a mercury thermometer, the reader should not underestimate the great difficulty which sometimes besets those who need highly precise determinations of temperature. Thus, while it usually suffices to determine the temperature of a stream of water by immersing a static thermometer in it, the temperature of a jet of steam cannot be so determined; rather, the thermometer would have to move with the jet (an obvious impossibility in the case of a high-speed steam jet), for temperature is a term which describes a certain property of matter in statistical equilibrium and really should not be applied at all to moving material. Likewise it is inapplicable to a single molecule.

When bodies reach a sufficiently high temperature they begin to emit visible light. They become *incandescent.* The radiation of visible light is thought to be the result of rearrangement of electrons within the atoms. If the effect of variable emissivities of the surfaces are eliminated it is found that all solids radiate light of the same tint if at the same temperature. Unless especially polished, most solids have nearly "black body emissivity"—and Table 1–1 applies with small variations.

TABLE 1–1. INCANDESCENT TEMPERATURES

Tint	Temperature
Dull red........	1100° F
Cherry.........	1300° F
Bright red......	1560° F
Yellow.........	1830° F
White..........	2102° F

Temperature is commonly expressed by *degrees,* a thermal degree being the unit measurement on an arbitrarily selected scale of molecular energy. The instrument for obtaining the measurement of temperature is a *thermometer* or a *pyrometer.* By common usage, the pyrometer is a device for measuring high temperatures.

The two scales most commonly used in thermometers are the *Fahrenheit* and *Centigrade.* A thermal degree on the Fahrenheit scale is $\frac{1}{180}$ of the change of thermometer index when the temperature of pure water which it is measuring is raised from that of melting ice to that of boiling water at standard atmospheric pressure. The Centigrade thermal degree is similarly estab-

lished except that it is $\frac{1}{100}$ of the same temperature range. The temperature of melting ice is made 32° on the Fahrenheit scale, and 0° on the Centigrade. This makes the boiling temperature 212° on the Fahrenheit, and 100° on the Centigrade scales.

A number of thermometers are shown in Figure 1–3. *A*, *B*, and *C* are glass stem thermometers holding a supply of mercury in a bulb at one end of the glass capillary tube. When the mercury is raised in temperature, volumetric expansion drives it upward into the capillary where the magnitude of its expansion can be visually observed. Instruments like those at *D* and *E* could be operated by either gas or vapor pressure generated in a bulb which is heated by contact with the fluid being measured. This pressure is conveyed through a tube joining bulb and instrument. The latter is fundamentally a pressure instrument which, however, is calibrated to read in thermal degrees. The temperatures of surfaces of solids can be taken sometimes by taping thermometer bulbs to the surface of the solid; otherwise thermocouples may be embedded in the solid. The thermocouple is a pair of electrical conductors of different materials permanently joined at one end. If the junction is exposed to some temperature by being embedded in a solid, immersed in a fluid, etc., thermoelectric phenomena create an electromotive force at the junction proportional to the temperature. The leads from the thermocouple are connected to what is essentially a potential galvanometer whose dial can be calibrated in thermal degrees.

Temperature measurement can be taken by immersion of the thermometer in the material, transferring by conduction the molecular energy level to the thermometer medium. It can also be measured at a distance by radiation effects—as in certain pyrometers.

An optical pyrometer, as the diagram shows, is one in which the eye compares the radiation emanating from an incandescent object whose temperature is to be measured, with that of a filament electrically heated in the tube of the pyrometer. By interposing between the eye and the hot body a red glass screen, monochromatic light is received by the eye. The current to the bulb is adjusted by rheostat until the intensity of emission from the filament exactly equals that from the light source. It is readily possible to make this adjustment, as with monochromatic light received both from the hot body and the filament, the filament appears black against the image of the hot body when the filament temperature is lower than that of the hot body, white when it is higher than that of the hot body, and disappears when the temperature is the same. In use, the pyrometer is pointed at the hot body, which is viewed through the eyepiece. The pyrometer is held in one hand, and the rheostat manipulated by the other until the filament disappears against the background. A sensitive ammeter, which measures the current flowing in the filament circuit, is calibrated directly in degrees of temperature.

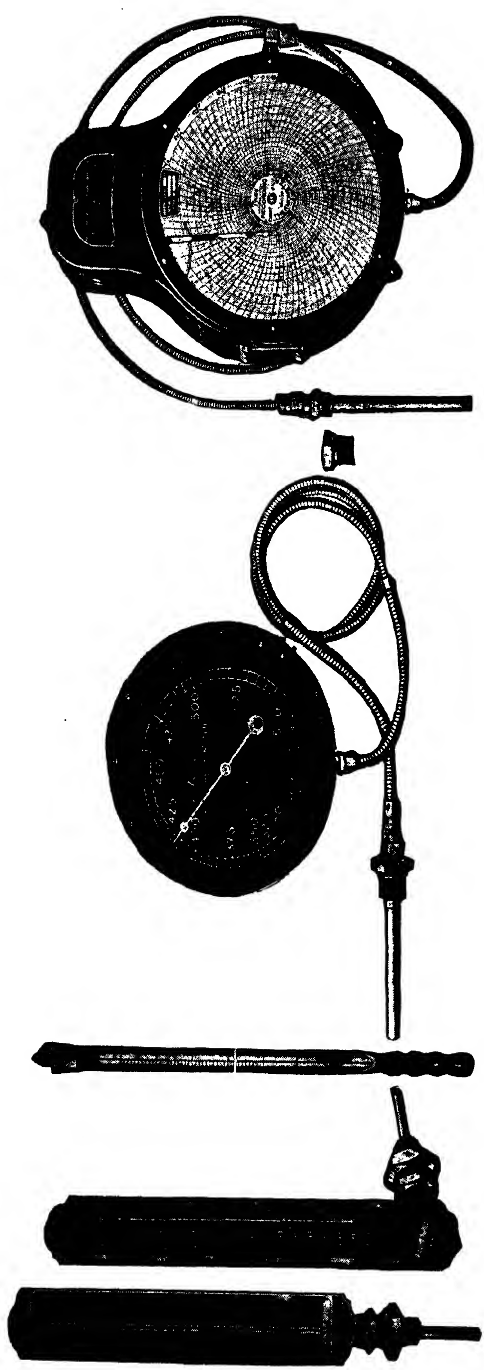
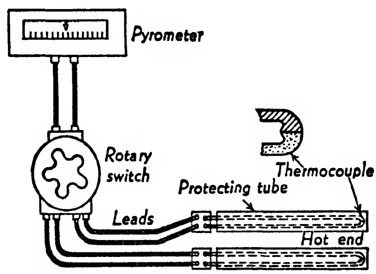
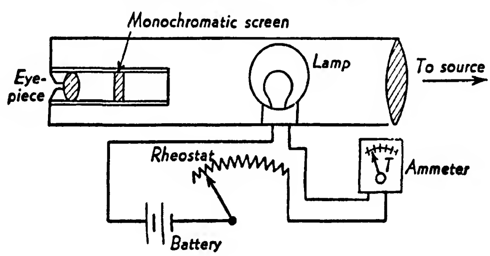


Fig. 1-3. Temperature measuring instruments.

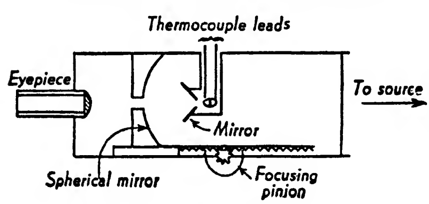
Total radiation pyrometers are equipped with a mirror which concentrates energy received by radiation from the source upon a thermocouple. This thermocouple is connected to a measuring instrument in a manner similar to that of the thermoelectric pyrometer. Since the total radiation received by a body at T_2 from one at T_1 is $K(T_1^4 - T_2^4)$, no great error is involved in high-temperature pyrometers by neglecting the term T_2^4. The



Thermocouple type (thermoelectric).



Leeds and Northrup type (optical).



Fery type (radiation).

FIG. 1-4. Pyrometers.

heat energy received, then, is proportional to the fourth power of temperature, so the galvanometer in the thermocouple circuit may be calibrated in degrees of temperature of the source which is radiating energy to the mirror, which converges it on the thermocouple.

Radiation and optical pyrometers have this advantage over the thermoelectric pyrometer; no part of them need come into direct contact with the hot body, and no part need be raised to a high temperature.

A series of substances having different fusion temperatures might serve roughly to measure the temperature of high-temperature regions such as furnaces, since, with a series of substances having progressively increasing

fusing temperatures, the temperature naturally lies between the fusion temperature of the last substance fused, and that of the next not fused. A series of artificially prepared mixtures, mostly of the oxides such as clays, lime, feldspar, have been designed to form a series of "Seger cones."

The Seger cones are triangular pyramids about $\frac{1}{2}$ inch on each side of the base, and 2 inches high. In use, the temperature is estimated, and four cones of consecutive serial numbers which are thought to include the temperature to be measured are placed on a refractory slab and inserted in the high-temperature region. If the cones have been properly selected, they will exhibit a range of behavior in the furnace varying from complete fusion of some to others remaining unaffected. One will soften until its tip bends over to touch the base, and that one is taken as indicating the temperature.

For regulating the temperature of fluids, thermostats are used. The thermostat is a device, the purpose of which is to regulate, automatically, the temperature of a body or of an enclosure, or to maintain the temperature at some predetermined value, or within a certain range.

Thermostats are actuated mainly either by the expansion of a fluid, or by expansion of a metallic element. A very common example of the former is the wafer-type thermostat, in which thin disk-shaped shells, called sylphons, are filled with gas, or partially filled with a liquid of suitable boiling temperature. Actuated by the developed internal pressure, the expansion and contraction of these sylphons, the cases of which are flexible, constitute a motion which can be mechanically transmitted to a regulating *valve* or *relay*.

Metallic element thermostats employ the well-known principle of linear expansion with temperature change. The element is not always built as a straight rod; satisfactory thermostats are constructed of a bimetallic strip wound in spiral form, which tends to coil or uncoil with changes of temperature.

Thermostats are sometimes actuated by thermoelectric currents or by the varying resistance of conductors, the effects of which, suitably amplified, may be utilized to control heating elements.

1-4. Pressure. Pressure is sometimes defined as the mechanical action between two bodies. When one of these bodies is composed of the molecules of a fluid and the other those of the confining walls, the action is fluid pressure. Fluid pressure acts equally in all directions (unlike compressive stress which has direction). This pressure is generally used as unit pressure, that is, force per unit area. Common units in the English system are pounds per square inch (psi.) and pounds per square foot (lbs. per sq. ft.). The height of a fluid column sustained by the pressure is sometimes used as a measurement of pressure. Mercury manometers and barometers and water manometers are examples. One inch of mercury (in. Hg) is equivalent to 0.491 psi. at ordinary temperatures while an inch of water (in. H_2O) is only 0.0361 psi. Atmospheric

pressure varies with time and place, but is often standardized at 14.7 psi., 29.92 in. Hg, 760 mm. Hg, 34 ft. H_2O, 1013.25 millibars. A millibar is a unit of atmospheric pressure frequently used in meteorological studies and is a unit pressure of 1000 dynes per sq. cm. Absolute zero pressure corresponds to a perfect vacuum and would exist in an enclosed region from which all fluid molecules had been removed. Most measuring devices determine the pressure with respect to the surrounding atmosphere, hence atmospheric pressure is of more than ordinary interest.

Barometers are used to measure the absolute pressure of the atmosphere, the standard being the mercurial barometer. It differs little from Torricelli's tube. Torricelli in 1643 utilized a column of water to measure atmospheric pressure. The modern barometer is a glass tube partially filled with mercury and evacuated at the closed end. As is shown in the accompanying figure, the tube is inverted in a reservoir of mercury exposed to atmospheric pressure. A brass measuring scale is fixed to the glass and is used with a sliding vernier for accurate readings.

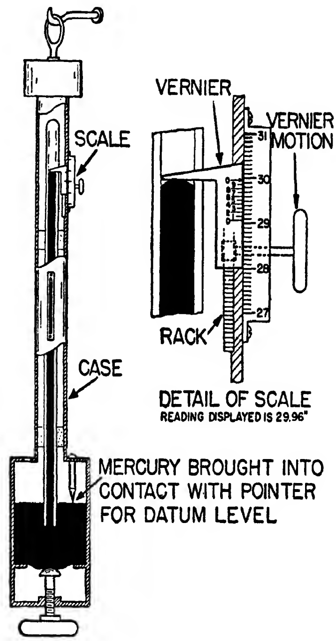


FIG. 1–5. Principle of Fortin-type barometer.

The Fortin-type instrument has a flexible section in the reservoir base so that regardless of temperature, the mercury level in the reservoir can be adjusted to a fixed point. A thermometer is affixed to the tube so that corrections for thermal expansion of mercury in the tube may be made if high precision of pressure reading is wanted. The standard barometer readings previously alluded to were for 32° F column. If the barometer reading at t_1° F is known and it is to be transferred to an equivalent reading for a t_2° F column, multiply the height at t_1 by $1 - 10^{-4}(t_1 - t_2)$. Mercurial barometers are not conveniently transported or read while in motion. A type suitable for this service is the aneroid barometer. Aneroid means "containing no liquid." This instrument depends upon the movement of a hollow, air-tight corrugated metal box, the inside of which is evacuated. Outside atmospheric pressure tends to collapse the box, collapse being resisted by spring pressure. Spring pressure does not vary; therefore, when atmospheric pressure varies, the sides of the box move slightly. This movement is multiplied by mechanical means to swing a pivoted pointer over a circular dial calibrated in equivalent in. Hg atmospheric pressure.

A gage is an instrument or device the purpose of which is to measure some physical characteristic such as length or pressure. The term is usually confined to mechanically activated apparatus, electrical measuring devices usually being called *instruments.* Pressure enters as a variable factor in a vast number of physical phenomena, and there is frequent need for its measurement over a very wide range. For ordinary pressures there are two gage types; viz., the familiar liquid manometer, of which the mercury barometer is an example, and those instruments dependent upon the deformation of a closed, thin-walled cell of elastic metal, such as the ordinary (Bourdon) steam gage and the aneroid barometer. Such gages are secondary instruments in that they must be calibrated in comparison with a primary gage, such as a manometer. Figure 1–6 shows the widely used Bourdon type of gage. An oval tube is bent in the form of a portion of a circle and sealed at one end. The other end is connected to a region whose pressure is to be measured. If that pressure is above atmospheric, the tube tends to change from oval to circular cross-section, accompanied by an uncurling action. This tendency is resisted by the elastic properties of the metal of the tube, but the tip executes a movement which is proportional to the change of internal pressure. If the internal pressure were below atmospheric, the effect would be reversed and the tube would tend to curve in, moving the sealed tip inward. The extent of this motion is quite small, and it must be amplified by suitable mechanism, so that the limited movement of the tip of the tube may be converted into a full swing of the indicating needle. Modification of the material of the tube, as well as the proportions and wall thickness, will adapt this instrument to a wide range of pressures.

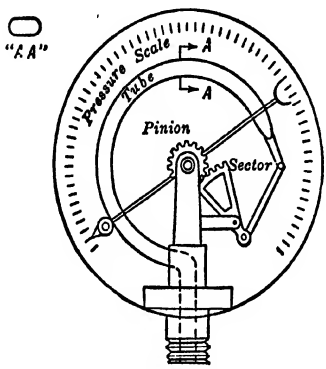


FIG. 1–6. Pressure gage (Bourdon tube).

Pressures slightly above atmospheric are known as plenums. Such pressures usually occur in air or gas systems as the result of the action of fans or blowers. The *plenum* is measured in small units of pressure, such as ounces per square inch, or in inches head of a liquid in a manometer. *Vacuum* is depression of pressure below atmospheric. The greater the vacuum the less the absolute pressure. Small vacuums are generally registered on water manometers; higher vacuums on mercury manometers or Bourdon-type gages. It is apparent from the above descriptions that absolute pressure of a high-pressure chamber to which a pressure gage is attached is gage pressure plus atmospheric pressure. The absolute pressure of a plenum is similarly plenum

plus atmospheric pressure, but in a vacuum the absolute pressure is atmospheric *less* the vacuum.

Pressures which are so small that the Bourdon tube *pressure gage* is not accurate, are conveniently measured by a liquid manometer. This instrument is one which balances fluid heads in a glass tube so that readings may be taken by comparing the registry of meniscii on a scale mounted alongside the tube. In its simple form, it consists of a U-tube, one end of which is open to the atmosphere, and the other to the region where the pressure is to be measured. If the pressure is different from atmospheric, the liquid with which the manometer is partially filled will stand higher in one leg of the tube than the other. As shown in the accompanying illustration, the

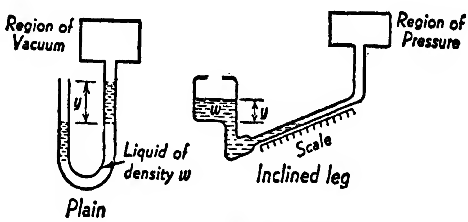


FIG. 1-7. Manometers.

manometer is connected to a vacuum, with the balancing head on the low-pressure leg. Since the other leg is open, the pressure head y is either the vacuum or the gage pressure. The magnitude is yw pounds per square foot when y is measured in feet and w in pounds per cubic foot. To make the instrument more sensitive, one leg may be inclined at a large angle to the other so that a given vertical displacement of the meniscus will travel a considerably larger distance along the scale. Of course in this type the other leg must be of enlarged cross-section so that its level will not vary appreciably as the meniscus travels along the scale. The scale can be calibrated to read any desired pressure units.

1-5. Volume. Volume is a measure of space. Since space is three-dimensional, the unit of volume is the cube of any linear dimension such as inches or feet. Cubic inches (cu. in.) and cubic feet (cu. ft.) are common units of volume employed in the designation of space occupied by solids, gases, and vapors. These also apply to the measurement of liquid volumes, although quarts, gallons, liters, etc., are customary units in liquid measure. Volumes of solids and liquids are readily measured, but those of gases and vapors offer more difficulty since direct linear measurement of physical size is impossible save when the dimensions of an enclosing surface may be employed for the purpose. An area moving in a direction normal to the plane of the area sweeps through a volume known as the displacement of that area. For example, a piston of A sq. in. face area moving a distance of L in. produces

a piston displacement of $A \times L$ cu. in. Heat power processes frequently include a change of volume (expansion or compression); or a rate of flow, which is the volume of a fluid passing a given station per unit time. Expansion describes an increase of volume of a substance, compression a decrease. Change of volume usually follows changes of pressure, temperature, or chemical composition.

The volume of a unit mass of a substance is a measure of some importance, being of the nature of a basic physical property of the substance. It is known as *specific volume,* the reciprocal of which is density. The specific volume of solids and liquids is little affected by changes of pressure or temperature, but gases and vapors are greatly affected by variations of these properties. Specific volume of these fluids can be expressed as a function of pressure and temperature either by equations or by tabulation in *tables of thermodynamic properties.*

1–6. Energy. Energy is undoubtedly the most important and most fundamental physical entity. It used to be defined frequently as "the capacity to do work." This is a point of view which considers energy as an intangible thing which a body receives when work is done on it or which it imparts when it does work on another body. Generally, mechanical work is implied. It is the product of the motion of a force and the component of the force in the direction of motion. The unit of work is therefore a product of force and distance, i.e., the foot-pound. In the c.g.s. system the corresponding unit is the centimeter-dyne, called an erg. A larger unit, 10^7 ergs, is called a Joule. A foot-pound is approximately 1.355 Joules. If one pushes against the rear of a cart with a force of 20 lbs. while the cart moves forward 100 ft., then 2000 ft. lbs. of work have been done. If the cart were stationary and the push exerted for the same time, one might experience an equal muscular fatigue but no mechanical work would be involved, since there was no motion. To lift a 100-lb. weight upwards 5 ft. requires 500 ft. lbs. of work, but to hold it at that level takes no work, though admittedly it could become fatiguing.

This concept of energy, as being merely the capacity to do work, was outmoded when in the nineteenth century Rumford, Davy, and Joule discovered the equivalence of heat and work. Later it was established that light and other radiations were forms of energy. Then modern research and theorization in atomic structure, and conclusions by Albert Einstein, of Theory of Relativity fame, produced the doctrine that energy and matter were related and interchangeable, and that the former principles of conservation of mass and energy, long held to be valid, were only approximately true. However, since a minute mass, if converted to energy, will produce an enormous quantity of the latter (in terms of its usage by mankind) the vast majority of theory, computation, and practice in heat power can continue to be based on the old laws of conservation of mass and energy.

The impact of these developments is to focus attention primarily on energy as the basic concept and to consider work, heat, light, and electricity as separate manifestations of energy. Energy, like mass, has inertia and gravitational attraction, and is no longer the imponderable, intangible quantity it was long considered to be.

Energy is thus an inclusive term, having variant forms. But all forms are energy and therefore may be transformed into one or another, and at an equivalence that is a function only of the units in which the variant form happens to be measured. The meaning of mechanical work has been previously stated. Whenever this mechanical energy is associated with motion it is a form known as kinetic energy. But the capacity to do work is also recognized as a potential quantity, exemplified by the pile driver weight poised ready to descend, or by the wound spring. The mechanical kinetic energy formed by a body of mass m slugs * in motion with a speed of v ft. per sec. is $\frac{1}{2}mv^2$ ft. lbs. The mechanical potential energy of a poised weight of m slugs mass which may descend through a height of h ft. is mgh ft. lbs., wherein g is the acceleration of gravity, i.e., 32.2 ft. per sec.^{2}

Heat can be visualized as the kinetic and potential energy of the molecules. This molecular theory of heat is a convenient hypothesis upon which to build an explanation of the nature of heat. It is called the *kinetic theory of heat.* It is capable of laboratory verification by numerous experiments which indirectly demonstrate the kinetic character of molecules. When a body receives heat which increases its molecular kinetic energy it rises in temperature. But heat can be received and stored in the potential energy form also. Vaporization and fusion are examples. Neither exhibits any change of temperature, but both require energy input to the body.

So, heat may be absorbed by a substance in several ways. It may be absorbed in the form of external work done when the heated substance expands against a pressure. It may be absorbed by increasing internal energy associated with the motion of the molecules. Again, heat may be absorbed by change of state of the substance, examples of which are the vaporizing of a liquid, the melting of a solid, etc. The total thermal energy possessed by a substance includes that present in these various forms. Heat content at the absolute zero temperature (minus 460° F, or minus 273° C) is zero. However, an arbitrarily taken datum for heat content often proves more valuable than the absolute heat content; thus, for example, steam tables which display the heat content of steam do so upon the assumed basis of zero heat content at 32° F.

The unit of energy commonly used in heat measurements is that amount which will raise a unit quantity of water one thermal degree. This is obviously a practical unit growing out of a need for a measuring unit, not from an understanding of the basic nature of heat energy. In the c.g.s. system the

* Unit of mass, commonly weight in 16 ÷ 32.2.

gram of water heated one degree Centigrade (at 16° C) is a *calorie* of heat, while in the English system a pound heated one degree Fahrenheit is a *British Thermal Unit* (B.t.u.). More exactly, a B.t.u. is $\frac{1}{180}$ of the heat absorbed by one pound of pure water when raised from the temperature of melting ice to that of boiling water under standard atmospheric pressure.

Energy exists in the electrical form when electrons are forced against or are urged along by an electric potential. Common units of quantity and potential are the coulomb and the volt, the energy product of which is the volt-coulomb. The quantity of electrons represented by a coulomb is such that one volt-coulomb is a Joule. A similar, though much smaller unit of energy, is the electron-volt, equal to the energy possessed by an electron after it has been accelerated by a potential difference of one volt. This unit is useful in the fields of electronics and atomic physics. An electron-volt = 1.591×10^{-19} Joule.

When mass is converted directly into energy, as in radioactivity, a gram of mass disintegrates into c^2 ergs of energy, where c is the speed of light, cm. per sec. This is equivalent to 66,300,000,000,000 ft. lbs. A subsequent chapter will extend the mutual equivalence of the various forms of energy.

The technique of measuring quantity of heat energy is called *calorimetry.* Apparatus devised to make such measurements are calorimeters. Often calorimeters depend on a rise of temperature of water, contained in the calorimeter, to show heat present. Other physical effects which find use in calorimetry are the vaporization of liquids and the melting of solids. The heat content of vapors, heating value of fuels, and specific heats of materials are examples of data discoverable through calorimetry.

1–7. Sources of Energy. The sources from which man can and has obtained energy are:

1. Fuels.
2. Flowing streams of water.
3. Winds.
4. Ocean tides and waves.
5. Terrestrial heat.
6. Solar rays.
7. Atomic disintegration.

The fuels are the most easily utilized source. It is fortunate that deposits of fuel are abundant. These are at present the principal source of commercially employed energy, and are the original source of the energy involved in all heat engines.

The hydraulic plant has a source of energy in a flowing stream or a waterfall. Good opportunities for harnessing this energy are scarce, as the sites

occur only through the existence of favorable topographic features, and nature has not been lavish with these.

Energy from winds has served man for many centuries, but the total amount captured from nature in this manner is small compared to other sources.

With the exception of terrestrial heat and atomic energy, all of the sources of energy may be traced indirectly to the sun. Evaporation of surface water to form rain clouds which continually replenish the flow of water in streams is accomplished by the heat of the sun; gravitational effects account for tides; warming and cooling of different portions of the earth's atmosphere cause winds and thereby waves; and solar rays, nourishing tropical vegetation through the prehistoric ages, may be held responsible for the deposits of coal, oil, and gas which were formed from that vegetation.

In a few instances the direct rays of the sun have been used to generate energy. The obvious fault of this source is that it is effective during the daylight hours only and, for continuous service, some reservoir of energy is necessary to carry through the night operation.

The suggestion of tremendous unleashed power in the daily silent glide of the tides and the evidences of gigantic terrestrial reservoirs of heat to be seen in geysers and volcanoes have, for years, prompted many dreams of harnessing these sources.

In 1945 a new type of bomb exploded over a Japanese city. It was a small bomb, but its concussion blasted the city. It released to the world the knowledge that science had at last succeeded in releasing, spontaneously, atomic energy. Although the details are yet secret, the enormous quantity of energy involved was only too apparent. The energy released was manifested mainly as heat available at temperatures heretofore unattained in the world. Great significance is attached to this new control of matter by man, as it may be possible to develop peaceful uses, including the development of energy for mechanical work. Except as a super blasting agent, uncontrolled explosive action would be of slight peaceful use. It remains to be seen whether engineering science can extend its control over atomic energy to the point where it may become a useful source of energy for heat power.

1-8. Thermodynamics. *Energy* has been shown to include a vast realm of physical study. This book will deal mainly with that phase of energy which includes the production and utilization of heat. Thermodynamics is that branch of the study of energy which treats of heat in motion, such as the conversion of heat energy into other forms, mainly the mechanical work form, and the transfer of heat from one medium to another. Its scope is usually extended to include the production of heat, and reversed conversions, i.e., mechanical work into heat as in refrigeration. Thermodynamics is the scientific background of theoretical and applied heat power technology.

The science of thermodynamics is usually set upon two postulated laws. The *first law* is called the law of conservation of energy. It states that during any thermodynamic change of a system, energy can neither be created nor destroyed, but the different forms (work, heat, etc.) are mutually interconvertible. When, in some thermodynamic change, a quantity of one form disappears, an equivalent quantity of another form must necessarily appear. Doubt is cast upon the absolute validity of this law by the modern concept of energy mentioned in a previous section. Until the spectacular release of vast quantities of energy by atomic fission, first accomplished in 1945 as a war weapon, no experimental evidence went contrary to the first law. Except where atomic energy release is concerned, it remains a valid basic law of thermodynamics. Even where chemical changes (combustion) occur, the heat liberated by the reaction could be considered potential energy prior to the change, and the idea of conservation of energy extended to include these cases. The simplicity of the first law is helpful. Often computation of certain heat quantities requires nothing more than an accounting (by simple addition and subtraction) of all energy units involved. The Heat Balance, as a useful tool in heat power analyses, will be mentioned again. Lest, in its simplicity, it be taken for granted as being obviously valid, the reader is asked to keep in mind that it rests squarely on the first law.

Now the *second law*, sometimes called the law of degradation of energy, is by no means as simple. It will be stated here, but the full import will be developed farther on.

One statement of this law reads: It is impossible for a self-acting machine to convey heat from a lower to a higher temperature. Another: Mechanical energy is a high-grade form, heat a low-grade form. High-grade has an irresistible tendency to go over into the low-grade form (as by friction), but not the reverse, for heat power plants are necessary to force that conversion. And then only a pitifully small part of the heat can be converted into high-grade energy. Still another form of the law is: No heat engine can continuously convert all the heat supplied to it into work. So the second law tells of the availability of heat for doing work and explains why some thermodynamic conversions are so much more readily accomplished than others, and why perpetual motion machines must remain a feature of "cloud cuckoo land."

1–9. Processes and Cycles. A fluid has a *state* determined by or associated with physical *properties*, especially pressure, volume, and temperature. If, after an interval of time some of these properties have undergone a change, a new state is possessed by the fluid. The change of state is called a *process*. The process may exhibit changes in all physical properties, or one may remain constant while others change. The latter condition is often found to exist.

The important processes of heat power are associated with changes taking place (1) in a fluid flowing steadily through a region, or (2) in a fluid confined in a certain region (non-flow).

A series of changes executed in orderly sequence, by means of which a mechanism, a working substance, or a system is caused periodically to return to the same initial condition, constitutes a *cycle*. Many complicated machines or assemblages of machines work in definite cycles. An important form of cycle is the heat engine cycle, in which a series of thermodynamic changes in a working medium periodically return the system to the same thermodynamic level. This working medium may be a gas, as in the Otto and Diesel cycles, or a vapor, as in the steam cycle.

1–10. Gas Processes. The expansion or compression of a gas undergoing a non-flow type of process may assume a variety of forms, depending on the extent to which heat is added to or rejected from the gas during the process, and also on the amount of mechanical work involved. There are, theoretically, an infinite number of possible expansions from an initial pressure P_1 and volume V_1 to a volume V_2. All these expansions could be represented on the PV plane by a family of curves $PV^n = C$. They are called *polytropic processes*. n may have any positive value from 0 to ∞. Having been numerically fixed, it determines the type of process. Four of these processes deserve special mention. Three of them are created by allowing a change of state during which pressure, temperature, or volume, respectively, are caused to remain unchanged. The fourth exists when the other three are variable but when no heat is directly transferred to or from the gas. These processes (in the order mentioned) are named isobaric, isothermal, isometric, and adiabatic.

If pressure is constant, the process is isobaric and $n = 0$.
If temperature is constant, the process is isothermal and $n = 1$.
If volume is constant, the process is isometric and $n = \infty$.
If transferred heat is zero, the process is adiabatic and $n = \gamma$, where γ is a certain thermodynamic constant which depends mainly on the kind of gas, i.e., hydrogen, air, etc.

The reader may secure some conception of how these processes could be carried out by reference to Figure 1–8.

These thermodynamic processes, as they occur in useful machines, are not often of the exact polytropic form desired. For example, an adiabatic process which is exemplified, at least theoretically, by expansion of the burned gases after the explosive combustion in the gasoline engine, is modified slightly by the interchange of heat between gases and cylinder wall, whereas a true

adiabatic has no heat either added or rejected in this way. The particular polytropic curve which would suit these conditions of expansion would depart somewhat from the adiabatic form.

During a polytropic process properties of the working medium are constantly varying, and analysis may be aimed at determining one of the following: the work done, the heat added, the variation of temperature. Some information may be obtained merely by comparing the value of the exponent n with certain other data. For example, if n lies between 0 and 1,

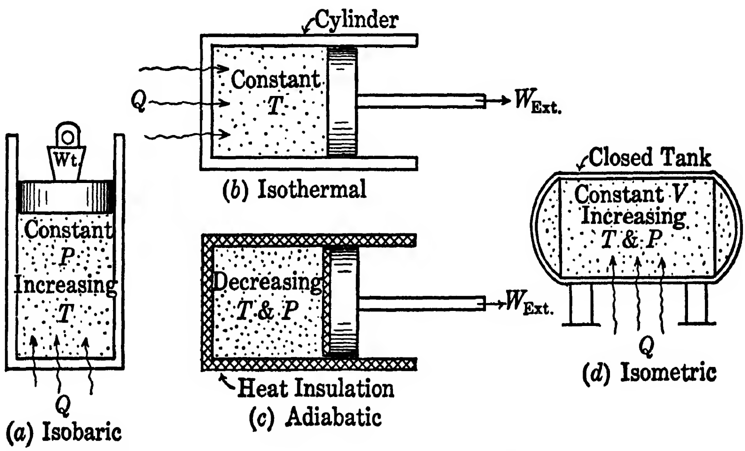


FIG. 1–8. Schematic arrangement of equipment for carrying out the four common gas processes.

NOTES:

a. When heat is added the volume expands, raising the piston. Work is done in increasing the potential energy of the weight (including piston) and in pushing back the surrounding atmosphere.
b. The work done is that of external work, $W_{\text{ext.}}$, plus the same displacement work against atmospheric pressure mentioned for the isobaric case. This gross work exactly equals the heat input when expressed in same units.
c. The internal energy of the gas must decrease in the amount of the gross work performed.
d. No work is done and the heat becomes an addition to the internal energy of the gas.

the temperature rises during an expansion and falls during a compression; when n is greater than 1, the temperature falls during expansion and rises during compression. Also, when n is less than γ, heat must be added to obtain an expansion whereas, when it is greater than γ, heat must be expelled. From the above it will be noted that there is a certain range of polytropic expansion in which, although heat is added, the temperature falls. This may seem to some to be paradoxical, but it is readily explained. During these expansions work is being done by the gas at a rate greater than that at which heat is being added, with the result that the deficiency must be made up from within the gas. The only way that this may be accomplished is for the gas to cool and give up some of its internal energy.

The equations for work done and for heat added in the case of polytropic expansion from state P_1, V_1 to state P_2, V_2 are:

$$W = \frac{P_1V_1 - P_2V_2}{n - 1}.$$

$$Q = (P_1V_1 - P_2V_2)\left(\frac{1}{n-1} - \frac{1}{\gamma - 1}\right).$$

Both of these are expressed in foot-pounds. Sometimes a substitution of a definite value of n in one or the other of these equations leads to an indeterminate: for example, with the isothermal,

$$W = \frac{P_1V_1 - P_2V_2}{1 - 1}.$$

But, since the equation of the isothermal for an ideal gas is $PV = C$,

$$P_1V_1 = P_2V_2,$$

and the work equation becomes indeterminate:

$$W = \frac{0}{0}.$$

The isothermal work equation, as deduced from another source, is

$$W = P_1V_1 \log_e\left(\frac{V_2}{V_1}\right).$$

Since the constancy of temperature requires that internal molecular energy remain unaltered, a quantity of heat energy equivalent to W must be applied to the gas during an isothermal expansion.

1-11. Gas Laws. By experimentation certain facts connected with gas processes have been discovered. Some of these are so important that they are fundamental to gas thermodynamics.

Boyle's Law. The English physicist Robert Boyle, while experimenting with air, discovered that the pressure varied inversely as the volume if temperature were held constant. This is the law of the isothermal process, for, if

$$\frac{P_1}{P_2} = \frac{V_2}{V_1},$$

then

$$P_1V_1 = P_2V_2 \quad (\text{also} = P_3V_3 = P_nV_n = \text{a constant}).$$

Charles' Law. Many years later Charles and Gay-Lussac, both Frenchmen, independently discovered a linear relation between volume and temper-

ature, when pressure is constant; likewise between pressure and temperature when the volume is constant. Their findings could appear (for an experiment on a tank of air being cooled) as shown in Figure 1–9.

Obviously, P has a straight-line-with-intercept relation with temperature in degrees Fahrenheit. If one adds the x intercept AO to $t°$ F and calls it T, then P is directly proportional to T, i.e.,

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}.$$

By cooling the gas to the lowest possible temperature, obtaining P and t data, then extending the plot of that data to intersect the t axis, the numerical magnitude of AO is discovered. Surprisingly, it is found to be the same for all gases. On the Fahrenheit scale point A is $-460°$, hence AO is 460° and $T = t + 460$. *Absolute zero* is the designation of temperature at A and the absolute scale of temperature, using the Fahrenheit thermal degree, is named "Rankine." Absolute temperatures are generally designated T, Fahrenheit temperatures t. Note that, wherever the proportionality of pressure and temperature, or volume and temperature (Charles' Law) is used (or any equation in whose derivation this law entered), temperature must be absolute.

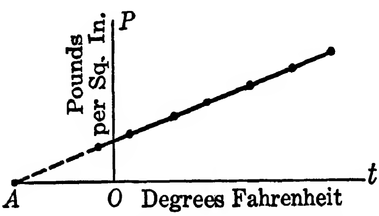


FIG. 1–9. Charles' Law graphed.

At absolute zero, molecular activity ceases. Bodies have no heat energy. The molecules have come to rest relative to each other.

Gases follow these laws closely—but not exactly. It is presumed that if a *perfect gas* were available it would follow the law exactly. To be "perfect," a gas should have no internal molecular friction, that is, molecular impacts should be perfectly elastic. The permanent gases nearly, but not exactly, achieve this and are commonly used as though they were perfect.

General Gas Law. It is noted that Boyle's Law covers the isothermal processes, Charles' the isobaric and isometric. To be valid for the adiabatic and other processes, a law should contemplate the simultaneous variation of P, V, and T. Assume that a sample of gas exists at a state described by the properties P_1, V_1, and T_1. Neither of the laws mentioned heretofore would cover a process which terminated at P_2, V_2, and T_2. But Charles' Law in two successive applications could cover the change to the P_2, V_2, T_2 state. With pressure constant the volume V_1 and temperature T_1 could be changed to an intermediate state, say V_a, T_a, such that $V_a = V_2$. Then a constant volume change could alter P_a $(= P_1)$ to P_2, accompanied by change of T_a to

T_2. A derivation following this scheme results in a general gas law, applicable to any process:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}.$$

As the term PV/T is a constant for any state, the above equation might be written

$$\frac{PV}{T} = C, \quad \text{or} \quad PV = CT.$$

Should it be desired to introduce the weight of the gas sample, C would be replaced by wR, where w is weight and R is a constant. R varies with the kind of gas, but is independent of pressure, volume, or temperature.

1–12. Vapor Processes. Vapors, being expansible like gases, may be used similarly in thermodynamic processes: isothermal, isobaric, etc. In addition, vapors may pass through processes unknown to gases, since they are capable of vaporization and condensation. Unlike gases, the physical properties of vapors behave somewhat erratically upon change of state, and simple equations similar to those of the two preceeding sections are inadequate. More consideration will be given to the thermodynamic properties of vapors (especially steam) before further discussion of vapor processes is begun.

1–13. Power. The *rate* of production or consumption of energy is called power. The unit is therefore the energy unit divided by a time unit. Thus: $\frac{\text{ft. lbs.}}{\text{sec.}}$, $\frac{\text{B.t.u.}}{\text{min.}}$, $\frac{\text{Joule}}{\text{sec.}}$ (called also a watt). Power is used chiefly to describe the rate of production or consumption of mechanical and electrical energy. The foot-pound per second is such a small unit of power that a larger unit, called horsepower (hp.), has been devised for common usage with heat engines. James Watt, in seeking for a means of expressing the power of steam engines and water wheels at the dawn of the industrial revolution brought about by the extensive use of machinery, turned to the horse as a familiar source of power and one in terms of which power values would be easily comprehended. It is said that he actually experimented with horses, using the best draft animals available, and that it was as a result of his observations that the horsepower has now become standardized as 550 *foot-pounds* of work per second. This is equivalent to about 746 *watts*.

The *kilowatt*, a unit of *power*, defined as one thousand watts, ordinarily serves as the commercial measure for electrical power. Electrical power of one kilowatt used steadily for one hour involves an energy consumption of one kilowatt-hour. A kilowatt is equal to about 1.34 hp.

Since a 100-hp. engine produces mechanical work at the rate of 100×550, or 55,000 ft. lbs. per sec., it could, if properly harnessed, overcome a resistance

of 5500 lbs. at a speed of 10 ft. per sec. Or again, a 5-hp. motor operating a hoist could raise a 275-lb. weight at a speed of 10 ft. per sec., or a 550-lb. weight at 5 ft. per sec.

1–14. Efficiency. Efficiency is a term of many meanings and much misuse. In its usual significance it expresses useful output as a percentage of input. An *efficiency*, including only the effect of losses arising from mechanical sources, mainly friction, would be mechanical efficiency. A machine such as a hoist has an efficiency expressible only as the mechanical efficiency. A steam engine has not only a mechanical efficiency which measures the friction and windage losses, but a thermal efficiency which is an entirely different quantity. The mechanical efficiency of a hoist is the output divided by the input. The former is the weight lifted multiplied by the height through which it is lifted. The input is the pull exerted on the hoist, multiplied by the distance through which that pull acts. The mechanical efficiency of a steam engine is the ratio of net power available at the pulley to power developed by the steam in the cylinder. The mechanical efficiency of a pump is the ratio of power expended on the water to that which the motivating source supplies to the pump. These are a few examples of mechanical efficiency, and might be multiplied endlessly since any moving machine incurs friction losses, and consequently has a mechanical efficiency of less than 100%.

Thermal efficiency is output, in heat units, divided by the heat supplied or chargeable. Thermal efficiency may partially define the operating condition of both a machine or a static piece of equipment. In the case of static equipment, if it is well insulated it may have very high thermal efficiencies. A machine which converts heat supplied into work output (the steam engine, Diesel engine, etc.) is always characterized by low thermal efficiencies because the conversion of the low-grade type of heat energy into high-grade energy of mechanical work is accomplished with considerable difficulty. The thermal efficiencies of the best prime movers today rarely exceed 35%, and are not found higher than 40% even when optimum conditions of loading, maintenance, and fuel employed are present. Thermal efficiency of a prime mover may be based on the output at the shaft per unit of heat supplied, or upon the electrical output in the case of a generator drive.

Many other efficiencies will appear at appropriate places in this book, for it is a common method of mathematically stating perfection of performance and design. However, one is cautioned against looking to efficiency as a *raison d'être* and objective of all technology, as often high efficiency is deliberately sacrificed to obtain some other feature such as compactness, low cost of construction, etc. Many persons guilty of fuzzy thinking may hope to terminate the same successfully by vague references to efficiency in general, doubtless using the same reasoning as those who once argued "the King can do no wrong." This fault should studiously be avoided.

1-15. Heat Power Plant. A power plant is a machine or an assembly of machines and equipment, the purpose of which is to convert energy from a dormant or useless state to a useful one. This usually takes the form of the conversion of the latent chemical energy of a fuel into mechanical or electrical energy, a conversion that is accomplished with difficulty, and then only at the expense of the loss of a major portion of the energy while it is in the form of heat.

Power being the rate at which mechanical or electrical energy is produced, one may regard a power plant as a factory for the production of a given commodity, energy, from fuel as a raw material, the rate of manufacturing being the power capacity of the plant as measured in horsepower or kilowatts.

The heat power plants may be classified as internal or external combustion types. The internal combustion power plant is a self-contained unit using the products of combustion as the working medium. Since it is compact, having all parts included in one machine or engine, and, since its efficiency is, on the average, higher than that of the external combustion cycle, it has been used for automotive service, and where compactness and mobility were desired.

Types of fuels which have been required by internal combustion engines have been more expensive than those which may be successfully employed in the external combustion cycle, and so there is large power-generating capacity installed in external combustion power plants such as the modern steam plant. A cross-section of a typical steam plant shows the complexity of the modern high-capacity, external combustion cycle. The plant shown in Figure 12-13 illustrates this. At the left is the coal-receiving station, at the right electrical transmission lines over which the converted energy is sent off cross-country. Coal is burned on mechanical stokers. Products of combustion after passing through the boilers and heat recovery equipment leave via a chimney. Condensing the turbine's exhaust steam requires enormous quantities of river water (note tunnels). The building is set off into boiler and turbo-generator rooms. Below are basements housing auxiliaries of which this type of plant employs many.

PROBLEMS *

1. Derive a simple formula for the conversion of Fahrenheit degrees to Centigrade.

2. Convert (a) 80 degrees F to degrees C; (b) 350 degrees C to degrees F; (c) −40 degrees F to degrees C.

3. What is the absolute temperature of air whose temperature is 100° F? 460° F?

4. The air pressure in a compressed air tank is 20 in. Hg gage. What is the absolute pressure in psi. (pounds per square inch)? In lbs. per sq. ft.? In inches of water? In feet of water?

* If not otherwise specified in this and subsequent problem groups, pressures are to be absolute, temperatures Fahrenheit.

5. The gage on a turbine condenser reads 28 in. Hg vacuum. Barometer 29.8 in. Hg. What is the absolute pressure in psi. and lbs. per sq. ft.?

6. There is a 2-in. water plenum in a blower duct. What is the absolute pressure in psi. and in inches of Hg? Barometer 29.95 in. Hg.

7. A boiler pressure gage reads 150 psi. What is the absolute steam pressure in the boiler if the barometer stands at 29.5 in. Hg?

8. If on a certain day the atmospheric pressure is 29.92 in. Hg at a temperature of 10° F, what would the barometer read in a boiler room where the temperature is 95° F?

9. A barometer reads 30 in. Hg on a certain day when the temperature is 32° F. What would the barometer read inside a freezing plant where the temperature is 4° F?

10. Suppose a U-tube manometer having water legs is attached to a tank filled with gas which displaces the water levels by 2 in. in the U-tube. What is the pressure in lbs. per sq. ft. in the tank? Water weighs 62.5 lbs. per cu. ft.

11. A certain gas exerts a gage pressure of 20 lbs. per sq. ft. The pressure head in a U-tube manometer is 3.84 in. What is the density of the liquid in the manometer?

12. How much air (volume) is there in a rectangular tank 30 ft. long, 20 ft. wide and 12 ft. deep? If the tank were filled with water, how many cubic feet would be needed? How many gallons?

13. What is the volume of a cylindrical pressure tank 8 ft. long by 60 in. in diameter?

14. The piston of a certain steam engine is 8 in. in diameter and 2 in. thick. What is the piston displacement in cubic inches if the piston moves $1\frac{1}{2}$ ft.?

15. The stroke of a certain steam engine is 18 in. What is the piston displacement if the diameter of the piston is 1 ft.?

16. In Problem 15, what is the volume of the steam when the piston is halfway down its stroke if there is a clearance space in the steam cylinder equal to 5% of the piston stroke?

17. If the velocity of a fluid in a 2-in. diameter pipe line is 5.0 ft. per sec., what is the rate of flow in cu. ft. per sec.?

18. If a pound of a liquid occupies 0.016 cu. ft., what is its specific volume? Its density?

19. What is the specific volume in cu. ft. per lb. of a liquid which weighs 0.1 lb. per cu. in.?

20. Suppose a man pushes on the rear of a cart with a force of 50 lbs. If the cart moves forward 100 ft. while he is pushing, what is the work done in foot-pounds? In Joules? In ergs?

21. A man is mowing his lawn. If the handle of the lawnmower makes an angle of 45° with the horizontal, what is the work done if the mower moves 50 ft. while the man is pushing on the handle with a force of 30 lbs.?

22. If a 2-lb. brick falls from the top of a building 700 ft. tall, how many foot-pounds of work have been done when the brick hits the ground? If the brick is lifted straight up to the top of the building again, how much work is done?

23. What is the mass of a body whose weight is 60 lbs.?

24. What is the potential energy represented if the body in Problem 23 were poised to fall through a height of 1500 ft.?

25. What is the kinetic energy of the body in Problem 24 the instant before it strikes the ground? Calculate its velocity.

26. What is the kinetic energy represented by a flow of water from a 2-in. nozzle at a velocity of 450 ft. per sec.?

27. Assume that you have a sample of air enclosed in a tank at 50 psi. pressure and 60° F. You are going to cool the contents of the tank and take a series of readings of P and t to determine the state. Plot the probable readings you would obtain by cooling the gas to $-50°$ F. 1 in. = 20 psi. 1 in. = 20° F.

28. In Problem 20, what is the volt-coulomb equivalent of the work done? What is the electron-volt equivalent of the work done?

29. If it were possible to disintegrate all the water in a full 250-cc. glass, how many foot-pounds of energy would be given off?

30. How many B.t.u. are required to heat water from 60° F to 175° F at atmospheric pressure? (per gal. and per cu. ft.)

31. In a constant pressure process (p = 150 psi. abs.) 15 lbs. of air change volume from 20 cu. ft. to 10 cu. ft. Find (a) work, (b) Q, transferred heat. [Refer to Figure 1–8(a).] ($\gamma = 1.4$.)

32. (a) What would be the work done in compressing isothermally 6 cu. ft. of air at 200 psi. abs. and 80° F to a pressure of 800 psi. abs. ($\gamma = 1.4$)? (b) Also find Q. [Refer to Figure 1–8(b).]

33. Find (a) the work and (b) the transferred heat, Q, for a constant volume process in which 10 cu. ft. of nitrogen ($\gamma = 1.4$) are heated so that the pressure changes from 50 psi. abs. to 185 psi. abs. [Refer to Figure 1–8(d).]

34. A gas at 225.3 psi. gage expands isothermally from 3.75 cu. ft. to 15 cu. ft. Find the work and the transferred heat. [Refer to Figure 1–8(b).]

35. In an adiabatic expansion, without flow, of an initial 8 cu. ft. of helium (γ = 1.659), p_1 = 80 psi. abs., p_2 = 30 psi. abs. Find work and transferred heat. [Refer to Figure 1–8(c).]

36. Five cubic feet of air are heated at constant volume so that the pressure changes from 40 psi. abs. to 200 psi. gage. Originally the temperature was 15° C. What is the final temperature in degrees F?

37. Two cubic feet of air at atmospheric pressure are changed at constant temperature until the volume is 0.8 cu. ft. Find the final pressure.

38. Ten cubic feet of a gas undergo a change from 80 psi. abs. and 75° F to 175 psi. abs. and 120° F. What is the final volume?

39. What is the weight of air in an automobile tire of 1 cu. ft. volume if the tire gage reads 30 psi. and the temperature is 85° F (R = 53.3)? If the temperature is raised to 110° F, how much air must be removed to keep the pressure at 30 psi.?

40. What horsepower is required to raise 500 gallons per minute of water through a height of 125 ft.? Water weighs 62.5 lbs. per cu. ft., 8.33 lbs. per gallon.

41. What horsepower is represented by a flow of water from a 2-in. nozzle at a velocity of 450 ft. per sec.?

42. A long shaft mounted in bearings receives 8500 hp. at a pulley and delivers 8200 hp. to a belt. The difference is lost in bearing friction. If the temperature is to remain constant, how much energy must be conducted away from the bearings?

43. How large a weight could be lifted by a hoist operated by a motor drawing 10 kw. if the weight is to be lifted at 18 ft. per sec.? Motor efficiency = 82%.

44. What is the efficiency of the system in Problem 42?

45. Ninety horsepower are given to a generator which delivers 54 kw. What is the efficiency?

46. Work Problem 43 assuming the hoist is only 90% efficient.

CHAPTER 2

Thermodynamic Properties

2-1. States of Matter. There are three states of matter, the gaseous, liquid, and solid. Pure elements can exhibit all three states (not simultaneously), as can also most pure inorganic compounds. Energy is the means to cause a change of state. Substances which are ordinarily solid at atmospheric temperatures can be liquefied, or even vaporized, with sufficiently high temperature. Likewise matter which is gaseous at ordinary temperatures can be made solid with sufficiently low temperature. Most substances may be made to assume any desired phase by proper regulation of temperature and pressure. Substances that have been liquid but have gasified through the addition of heat are said to be *vaporous* if near the vaporization temperature. Vapor state is a "sub-layer" of the gaseous state. To distinguish between true gases and vapors, a vapor is defined as a substance in the gaseous state but below the *critical temperature*, which is the temperature at which liquid and vapor are the same because of high pressure.

Energy added to ice will cause a rise of temperature, as the molecules increase in kinetic energy. At some temperature which is determined by the pressure, the kinetic energy intensity has risen until further energy additions cause a loosening of the molecular binding forces which have restricted the molecules to a limited mean position. They are then visualized as wandering but meeting with frequent collisions with neighboring molecules. A considerable amount of energy is needed to accomplish this physical change, but the intensity of molecular energy represented by the temperature is not altered during the process. Obviously the energy is stored as a potential energy of the molecules. The process described is *fusion*, commonly called melting. Now, at constant pressure, imagine the liquid to continue to receive heat energy. The molecular kinetic energy is again augmented, and temperature rises until molecular activity is so violent that molecules can leave the surface of the liquid and remain in the region above it. This is *vaporization*, also called boiling. In the vapor state the temperature is the same as the liquid state immediately prior to vaporization, so the molecular kinetic energy is unaltered during the change of state. However, the average molecular spacing is much greater than in the liquid (at least until close approach to the critical pressure). The molecular spread represents a store of potential energy whose magnitude is quite large compared to the energy required to bring the liquid

from the melting to the vaporization phase. The potential energy increments mentioned are known as "energy of disgregation" and are called, respectively, heat of fusion and heat of vaporization. The energy causing temperature rise is called "*sensible heat*" since human senses perceive temperature.

The influence of pressure and temperature upon the state of matter is shown for a typical pure inorganic substance in Figure 2-1 where the boundaries of the different phases are shown by lines. Notice that the liquid phase does not exist below the triple point. Were the temperature of a solid increased at constant pressure P_1, it would pass directly from the solid to the vapor phase. This is known as *sublimation* but, if the pressure were P_2, then the solid state would go by fusion into the liquid and the liquid pass by vaporization to the vapor state.

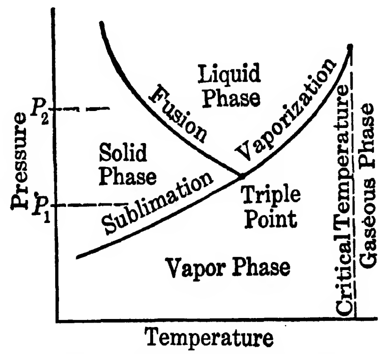


FIG. 2-1. Possible states of matter.

2-2. Specific Heat. Sensible heat is approximately proportional to the temperature increase it produces. The factor of proportionality, when expressed for a unit quantity of the substance, is specific heat. It is the heat units necessary to accomplish a degree temperature change in a unit quantity such as a pound, a cubic foot, or a mol.* It varies not only with the substance, but with (1) the phase of matter the substance is in, (2) the temperature of that phase, and (3) the type of process that the energy change is creating. The two most important gaseous processes for which specific heat is recognized are the isobaric and isometric. Isobaric specific heat will be represented by c_p; isometric by c_v. c_p of a gas is always larger than c_v because the same amount of internal energy must be added to the molecules per degree temperature rise in the isobaric as in the isometric case (see Figure 1-8), and in addition the isobaric expansion requires some heat energy to compensate for the external work done. The ratio c_p/c_v will be represented by γ. This is one of the important thermodynamic "dimensions" of a gas.

The specific heat of solids and liquids has only small variation with temperature and is little affected by pressure changes. But the specific heats of vapors and gases are quite variable and, in the case of vapors, somewhat erratically so. Tabulations of specific heat are available for the common liquids and vapors, while tables or equations of empirical nature give the specific heats of gases. Most reference books on thermodynamics contain extensive tabulations of such data.

* That quantity of a substance whose weight in pounds equals its molecular number.

TABLE 2–1. SOME PROPERTIES OF COMMON SUBSTANCES

Solids and Liquids

Name	Symbol	Molecular Weight	Density, lb./cu. ft.	Specific Heat,† B.t.u. per lb. per deg. F	Coefficient of Linear Expansion † per deg. F
Aluminum	Al	27	165	.207	$.128 \times 10^{-4}$
Brass	*	*	540	.088	$.104 \times 10^{-4}$
Concrete	*	*	160	.156	$.080 \times 10^{-4}$
Glass	*	*	160	.190	$.02 \times 10^{-4}$ to $.05 \times 10^{-4}$
Ice	H_2O	18	56	.504	$.45 \times 10^{-4}$ §
Iron	Fe	55.8	450	.101	$.065 \times 10^{-4}$
Magnesium	Mg	24.3	109	.241	$.142 \times 10^{-4}$
Mercury	Hg	200.6	848	.033	1.0×10^{-4} ‡
Petroleum	*	*	55	.50	4.0×10^{-4} ‡
Steel	*	*	490	.10	$.06 \times 10^{-4}$
Water	H_2O	18	62.4	1.0	3.0×10^{-4} ‡
Wood	*	*	30 to 50	.6	$.03 \times 10^{-4}$ to $.05 \times 10^{-4}$

Gases and Vapors

Name	Symbol	Molecular Weight	Gas Constant R	Av. Specific Heat,† B.t.u. per lb. per deg. F		γ †	Critical Temperature, deg. F
				c_v	c_p		
Acetylene	C_2H_2	26	58.8	.32	.39	1.24	97
Air	*	28.8	53.4	.17	.24	1.4	−220.3
Ammonia	NH_3	17	89.5	.42	.52	1.25	270.3
Carbon Dioxide	CO_2	44	34.9	.16	.20	1.29	88.0
Carbon Monoxide	CO	28	55.2	.18	.25	1.40	−218.2
Helium	He	4	386.5	.75	1.25	1.66	−450.2
Hydrogen	H_2	2	767.5	2.48	3.46	1.46	−399.8
Mercury	Hg	200.6	....	.015	.025	1.67	2800+
Methane	CH_4	16	96.4	.4	.5	1.3	−116.5
Methyl Chloride	CH_3Cl	50.5	30.6	...	.23	....	289.6
Nitrogen	N_2	28	55.1	.18	.25	1.39	−232.8
Oxygen	O_2	32	48.3	.16	.22	1.40	−181.8
Steam	H_2O	18	86	...	.5 up	1.6 to 1.8	705.4
Sulfur Dioxide	SO_2	64	23.6	.16	.19	1.20	315.0

* Mixture, or variable composition.
† For moderate temperatures.
‡ Volumetric coefficient.
§ At about 32° F.

2–3. Enthalpy. The enthalpy of a substance is defined as the sum of its internal and potential energy above some datum. Let h = enthalpy, u = internal energy, and PV the external potential.

$$h = u + \frac{PV}{J^*}.$$

A theoretical datum would be absolute zero of temperature. However, enthalpy tables for the practical use of engineers and others have typically a datum of 32° F for steam, −40° F for refrigerants, and 80° F for gases.

The incremental change of enthalpy for all gas processes is $c_p \Delta t$ where c_p is the isobaric specific heat and Δt † is temperature change. This is not true of vapor processes unless above the critical temperature.

2–4. Entropy. In the mathematics of thermodynamics a certain term appears so frequently that it is considered of equal importance with pressure, volume, and temperature as a variable defining the physical state of a substance. It is transferred heat divided by the absolute temperature of the substance, and is called entropy. The symbol s will be used for entropy. For example, the change of entropy of water during vaporization at atmospheric pressure is $\frac{970.3}{212 + 460} = 1.4446$ B.t.u./degree per lb.‡ Transferred heat, ΔQ, which is received by a substance as sensible heat, causes a change of temperature. In this case the increment of entropy change $\Delta s = \Delta Q/T$. For exact change of entropy between finite temperature limits the increment must be infinitesimally small, and the methods of integral calculus used to summarize it.

If there is no transferred heat, as is the case in an adiabatic expansion of gas, the numerator of the entropy term is zero and likewise the change of entropy itself. Then entropy becomes the constant property for adiabatic gas processes; hence isentropic process is a synonymous expression. Vapors can pass through isentropic processes also. Not all adiabatic changes are isentropic, but most of them are. All reversible adiabatic processes are isentropic.

Entropy also represents the organization of energy. Minimum entropy is maximum organization. In general, increases of entropy accompany disorganization and decrease of availability of heat for conversion into work. Friction, flow around or over obstructions, leakage through clearances or packings—these are typical entropy-increasing processes. It is theorized that in the beginning of time all energy in the Universe was concentrated in

* J is a symbol for the mechanical work equivalent of heat. See Chapter 4.

† The symbol Δ is used to denote an increment of the quantity to which it is attached. It will be used in this sense, and in no other, throughout this book.

‡ See Table 2–3.

one well organized mass. As time rolls on available energy tends to become unavailable; mechanical work is converted to heat, and energy increases in dispersion and disorganization. One property which stands out prominently in this dissipation process is entropy. It is always on the increase. The physicist Eddington called it "Time's Arrow," and said that "Nothing in the statistics of an assemblage can distinguish a direction of time when entropy fails to distinguish it."

2-5. Air. The term air is frequently used as synonymous with "atmosphere of the earth" and it is in that sense that it is used here.

Most of our knowledge of the composition of air is based upon samples that have been taken at the surface of the earth. There air is composed chiefly of oxygen and nitrogen. Air also contains a variable proportion of water vapor and small quantities of other gases. In many engineering calculations, air is considered to be composed only of nitrogen and oxygen, and the proportions assumed for such calculations are, by weight, 76.8% nitrogen and 23.2% oxygen; and, by volume, 79.1% nitrogen and 20.9% oxygen. A more detailed analysis of air (without taking into consideration the variable amount of water vapor) is given below:

AVERAGE COMPOSITION OF AIR *

(At, or near to, the surface of the earth)

Substance	Percentage (By Volume of Dry Air)	Percentage (By Weight of Dry Air)
Nitrogen	78.09	75.54
Oxygen	20.93	23.14
Argon	0.93	1.27
Carbon Dioxide	0.03	0.05
Neon	0.0018	0.0012
Helium	0.0005	0.00007
Krypton	0.0001	0.0003
Hydrogen	0.00005	0.000004
Xenon	0.000008	3.6×10^{-5}
Ozone	0.00005	1.7×10^{-6}

* Paneth, *Quart. J. R. Met. Soc.* **63**, 346 (1937).

Carbon dioxide is produced by the combustion of carbon-containing fuels and from the decay of organic matter. Its concentration in the air would be much greater if it were not consumed by vegetation in the process of photosynthesis, by which oxygen is liberated. In this way, the steady addition to the atmosphere of carbon dioxide by the combustion of fuels, and by the breathing of men and animals, is counteracted. and the oxygen-carbon dioxide ratio of the air is maintained.

The content of water vapor in the atmosphere varies greatly in amount, depending upon the locality, the season of the year, and the hour of the day, due to local and general states of the weather.

The standard density of air at 32° F and 14.7 lbs. pressure is .081 lb. per cu. ft. Its gas constant is 53.4 ft. per °F, and its composite molecular weight is 28.84. Above sea level the temperature of the atmosphere decreases 3.57° for each thousand feet of altitude up to the tropopause. The pressure decreases with increasing altitude as is shown in Table 2.II.

TABLE 2–2. VARIATION OF AIR PRESSURE WITH ALTITUDE

Altitude (Feet)	Pressure (Inches of Mercury)	Pressure (Pounds per Square Inch)
Sea Level	29.92	14.7
1,000	28.86	14.2
5,000	24.89	12.2
10,000	20.58	10.1
15,000	16.88	8.3
20,000	13.75	6.8
25,000	11.10	5.4
30,000	8.88	4.4
....	....	...
50,000	3.44	1.7

2–6. Gases. Air is a mechanical mixture of gases. As was stated in the preceding section, it is frequently assumed to be a mixture of oxygen and nitrogen. Other gases of interest to the heat power field would be gaseous fuels, such as methane (CH_4), and gaseous products of combustion.

Avogadro's celebrated enunciation of the theorem that equal volumes of gases at the same pressure and temperature contain equal numbers of molecules is sufficient authority for the principle of equal molal volumes. At the same pressure and temperature a mol of any gas occupies the same volume. This volume is approximately 380 cu. ft. per pound-molecular-weight at 14.7 psi. and 60° F. Thus the volume of 28 lbs. of nitrogen, 32 lbs. of oxygen, 44 lbs. of carbon dioxide, etc., is 380 cu. ft. under standard conditions. The molal volume for any other pressure p psi. and temperature T deg. Rankine is:

$$V = 380\frac{14.7}{p} \times \frac{T}{520^\circ} \text{ cu. ft.}$$

The gas constant R of the general gas law when multiplied by the molecular weight of the gas is nearly 1544 in the English system. This number is called the universal gas constant.

When w lbs. of a gas are heated through a temperature range the internal energy absorbed is the same as the transferred heat of an isometric process.

Consequently, internal energy change $\Delta u = wc_v\Delta T$ for any process. The enthalpy change, Δh, can be derived by thermodynamic reasoning from the definition of enthalpy, and is found to be $\Delta h = wc_p\Delta T$ for all polytropic gas processes. Similarly, change of entropy $\Delta s = \Delta Q/T$. Many of the thermodynamic properties of gases will be found in Table 2.I.

Example: Illustrating the calculation of state and process quantities for a typical gas, a constant pressure (isobaric) process is here computed. Parallel solutions might be obtained for other processes such as the isothermal. Assume 0.125 lb. of nitrogen in a constant pressure chamber at 22 psi. and 60° F. The temperature will be increased to 190° F by transfer of heat energy from external sources into the nitrogen. To analyze this thermal change, the original volume is first determined from the general gas law $PV = wRT$. A solution for V_1 gives $V_1 = wRT_1/P_1$. The numerical value of R will be found in Table 2-1. A comparison of the temperature of this nitrogen with the critical temperature of nitrogen demonstrates that the nitrogen is in a gaseous state and may be expected to obey, closely, the ideal gas laws. Making the necessary numerical substitutions for V_1,

$$V_1 = \frac{0.125 \times 55.1(60 + 460)}{22 \times 144} = 1.13 \text{ cu. ft.}$$

Now this volume will increase at constant pressure at a rate proportional to the absolute temperature change. This is in accordance with Charles' Law. Therefore, the final volume V_2 is obtained from V_1 by

$$V_2 = V_1\left(\frac{T_2}{T_1}\right).$$

Substituting numerical values in this equation,

$$V_2 = 1.13\left(\frac{190 + 460}{60 + 460}\right) = 1.41 \text{ cu. ft.}$$

The gas density is readily obtained from the foregoing, as is also the specific volume which is simply the reciprocal of the density. Thus, at the initial state

$$\text{Density} = \frac{0.125}{1.13} = .1107 \text{ lb. per cu. ft.}$$

and

$$\text{Specific volume} = 9.03 \text{ cu. ft. per lb.}$$

while, finally,

$$\text{Density} = \frac{0.125}{1.41} = .0886 \text{ lb. per cu. ft.}$$

and

$$\text{Specific volume} = 11.28 \text{ cu. ft. per lb.}$$

The work done by the constant pressure P lbs. per sq. ft. in an expansion of ΔV cu. ft. will be $P\Delta V$ ft. lbs. So, in this case, work done, $W = 22 \times 144(1.41 - 1.13) = 887$ ft. lbs.

The heat transferred into the nitrogen is $wc_p\Delta T$. In numbers, $Q = 0.125 \times 0.25(190 - 60) = 4.06$ B.t.u. The increase of internal energy $\Delta u = wc_v\Delta T$.

$$\Delta u = 0.125 \times 0.18(190 - 60) = 2.92 \text{ B.t.u.}$$

As increase of enthalpy during *any* gas process is always $wc_p\Delta T$, the enthalpy change for the isobaric process equals the transferred heat. Hence, $\Delta h = 4.06$ B.t.u. The change of entropy equals the summation of $\frac{\Delta Q}{T}$ but, since $\Delta Q = wc_p\Delta T$,

$$\Delta s = wc_p \sum \frac{\Delta T}{T} = 0.125 \times 0.25 \sum \frac{\Delta T}{T}.$$

By mathematical methods of the integral calculus it can be determined that the summation of $\frac{\Delta T}{T}$ between temperature limits T_1 and T_2 is $\log_e \frac{T_2}{T_1}$,

$$\Delta s = 0.125 \times 0.25 \log_e \left(\frac{190 + 460}{60 + 460} \right) = .00697 \text{ B.t.u. per deg.}$$

Had the heating occurred at constant volume instead of constant pressure, transferred heat would equal the internal energy increment (since no work would be possible), the Δh equation would incorporate c_p as before, and c_v would appear in the equation for Δs.

2–7. Vapors. A substance in its gaseous state but below its critical temperature is a vapor. Vapors are not subject to the laws of gases. The word "saturation" has a special meaning in this field. Liquid, which has absorbed the maximum quantity of heat possible while remaining in the liquid phase, and has, consequently, risen in temperature to the boiling point, is said to be "saturated." Here the meaning is "saturated with heat." Saturation temperature is synonymous with boiling temperature. The quantity of heat contained, as well as the boiling temperature, is a function of the pressure existing on the surface of the liquid.

A vapor whose temperature corresponds to the boiling temperature at the pressure existing on it, is said to be *saturated.* Expressing the same thought another way, a vapor is saturated when its temperature is a function of its pressure alone. A saturated vapor may be wet or dry, and the term does not imply, necessarily, a wet vapor. A vapor of 100% *quality* having no superheat, is said to be "dry and saturated." Quality describes the condition of a saturated vapor. A vapor in a condition intermediate between liquid and a dry vapor is said to have a certain quality which may be defined as the ratio of the vaporized portion to the total weight of liquid and vapor. The vaporization of a liquid requires the expenditure upon it of the latent *heat of vaporization.* When heat to this amount is added, the liquid is converted to a dry vapor. If x% of this heat is added, only x% of the liquid is vaporized. x is the quality. It is the per cent of dryness of a wet vapor. In contrast to a saturated vapor, the temperature of a superheated vapor depends both on the pressure and the degree of superheat. By virtue of the importance of water in both its liquid and vapor phases, its properties have been completely investigated and recorded in tables and charts. The properties of saturated

steam will be found among such compilations. The physical attributes of saturated steam are the pressure, temperature, volume, enthalpy, and entropy. These are always given for steam which is dry and saturated, leaving the reader to apply the quality factor when it occurs. The increase of volume on vaporization, and the latent heat of evaporation, are present in wet steam to the extent of the per cent dryness of the steam. One of the important entries in the saturated steam table is that for atmospheric pressure. At 14.7 lbs. per sq. in. absolute pressure, the saturation temperature of steam is 212° F. The heat contained in it as a boiling liquid is 180 B.t.u. (above 32° F), and its latent heat of evaporation is 970.3 B.t.u. per lb.

The evaporation of a given mass of any liquid requires a definite quantity of heat, dependent upon the liquid and upon the temperature at which it evaporates. The quantity required per unit mass at a fixed temperature is called the heat of vaporization of the substance at that temperature. It may be measured by allowing the vapor to condense in a suitable *calorimeter*, the heat thus evolved, corrected for fall of temperature before and after condensation, being observed. (The heat evolved in condensing is equal to that absorbed when the liquid evaporates.) The result is often surprising. For example, the evaporation of water at the boiling point requires about 970 B.t.u. per lb., or more than five times the heat required to raise its temperature from freezing to boiling. The explanation is the large amount of energy necessary to separate the molecules against their cohesion.

Superheat is the addition of heat to produce vapor at a higher temperature than saturation. Superheat is possible when the vapor is led away from the liquid from which it was boiled. The temperature added is called the degree of superheat, and the equipment to superheat is known as a superheater.

The effectiveness with which water vapor may be employed as a working medium in a power cycle is enhanced by superheating it. The less erosive character of dry steam and the lower heat losses from pipes carrying dry steam, have made superheating very desirable, so that most steam boilers are equipped with superheaters at present.

Vapors are employed as the working media for power cycles and for reversed power cycles (i.e., refrigeration cycles). The principal recommendation of vapors for these services is their heat-absorbing, heat-carrying, and heat-releasing ability. Large quantities of heat may be absorbed or released by vaporization or condensation. These are pure physical changes and require no change in chemical composition.

2–8. Steam. Steam is the vapor of water at or above its boiling temperature. The universal occurrence of water, its cheapness, its solvent properties, its suitability as a medium for a vapor cycle, and general familiarity with it and its properties, all have contributed to make of steam the most

important vapor used by man. If a closed vessel is partially filled with water, and heat applied at a high temperature, the water will absorb the heat and rise in temperature until its molecular activity (which depends on its temperature) is so increased that the internal molecular attractions are no longer able to maintain it in the form of a liquid, and molecules leave the surface of the liquid and occupy the space above it in the rarefied condition of a vapor. Water has a definite vapor pressure at every temperature; when the vapor pressure becomes equal to the pressure above the liquid, the boiling point has been reached. If the steam is produced in a closed vessel, it is invisible. If it is released to the atmosphere, it partly condenses to form a cloud of mist (small drops of liquid water), which is visible. The temperature at which the boiling takes place is dependent upon the pressure which is maintained in the boiler.

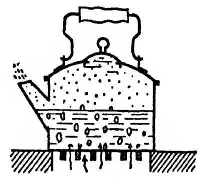


FIG. 2-2. Elementary boiler.

A steam boiler of very elementary character is pictured in Figure 2-2. Suppose one begins with a pound of water in the kettle at 32°. This temperature is frequently assumed as "datum" for the properties of water. The condition of the water and its changes upon subsequent additions of heat may be conveniently studied by graphing on a temperature-enthalpy plane such as Figure 2-3. So one begins with one pound of water at A. Heat is added through the bottom of the kettle. The water rises in temperature rather steadily as heat is added. The temperature rise would be directly proportional to enthalpy were specific heat absolutely constant. When one pound of water has finally reached a temperature of 212°, vaporization starts and temperature remains steady at 212°. The state at the inception of vaporization is that of "saturated water," and the temperature is the "saturation temperature." The heat required to effect the change from A to B is called *heat of the liquid.* As heat continues to flow into the water more of it vaporizes and the liquid level sinks until finally all of the water has been converted into steam.

From the water there has now been produced a pound of saturated steam. If this steam contains no droplets of unevaporated liquid it is "dry and saturated." Its condition is marked C on the graph. The *latent heat of vaporization* is that required to change the state from B to C. Now, if this steam could all be collected in a dry kettle at atmospheric pressure and heated further, the temperature could be made to rise to some point D, the exact position of which depends only on the quantity of heat that was added. The rise of temperature over saturation is the "degree of superheat," and the increase of enthalpy between C and D is the "superheat." Note that the enthalpy at B and C is a function of the pressure only; but at D it depends on both pressure and temperature. That is, saturated liquid or dry saturated steam

enthalpy (also volume, entropy, and temperature) is a function of pressure only, whereas enthalpy (volume and entropy, also) of superheated steam is a function of two independent variables—pressure and temperature. The behavior of water undergoing physical change could equally well be shown on such planes as T–s, h–s, P–V, etc., but t–h is especially useful in introductory studies of the thermodynamic properties of water substance. The other planes have their special uses, some of which will appear subsequently.

The saturation temperature of steam increases under pressure, at first rapidly, then more slowly, with uniform increments of pressure, until a temperature of 705.4° F is reached, at a pressure of 3206 lbs. per sq. in. The heat

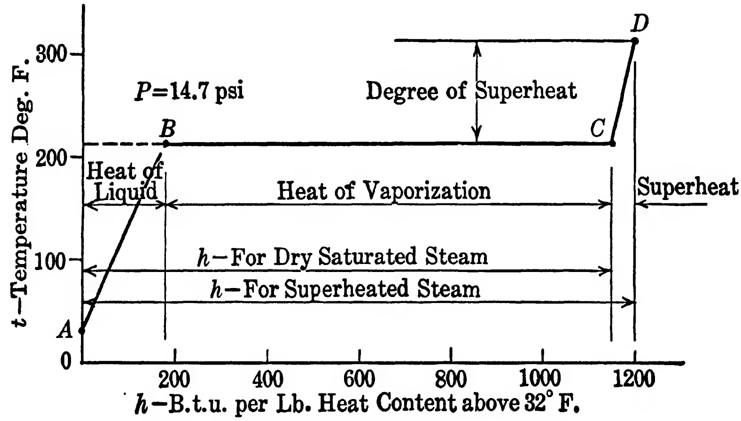


FIG. 2–3. Temperature-enthalpy variation during conversion of water into steam.

required to boil off a pound of water at the saturation temperature into dry saturated steam is greater at the lower pressures, and decreases as pressures increase, until it disappears at 3206 lbs. per sq. in. This high pressure is the critical pressure at which steam and water are identical in physical properties. Steam at a temperature above 705.4° F is certain to be superheated, no matter what the pressure. The heat contained in saturated water increases with increase of pressure until it becomes equal to the full enthalpy at the critical pressure. The rate of increase of the heat of a liquid does not correspond with the rate of decrease of the heat of evaporation, and their sum, that is, the enthalpy of the dry steam, increases until it becomes maximum between 400 and 450 lbs. per sq. in., after which it decreases. At the critical pressure a pound of saturated steam contains less heat than at any other pressure.

The open kettle design is unnecessarily bulky and inefficient as a producer of steam in quantity. Furthermore, steam is generally employed at pressures higher than atmospheric, hence a closed vessel is in order. In Figure 2–4 a simple pressure steam boiler has been developed by enclosing some heating flues in a vertical cylindrical steel shell so that most of the flue exterior

surface is submerged in water. A fire is built on the grates. Flame and hot gases rise through the flues to the chimney above. This keeps the flues hot and they naturally transmit quantities of heat energy to the surrounding water, resulting in rapid conversion of the water into steam. Bubbles of steam rise through the water and accumulate in the region above. If the steam outlet is closed the vapor accumulates at increasing pressure as the saturation temperature of the liquid itself rises. If at some desired operating pressure, say 50 psi., the valve in the steam outlet line is opened sufficiently to permit the passage of steam from the boiler as rapidly as it is generated, the pressure will remain constant. Finally, the addition of water (called feed water) at the lower connection in a quantity equivalent to the weight of steam discharged will maintain the water level at the desired height in the boiler, and steady continuous operation is possible. In actual boilers the steam is held to the desired pressure by varying the fire rather than by adjusting the outlet valve. That valve is opened wide and the steam flow is determined by the demands of the equipment connected to

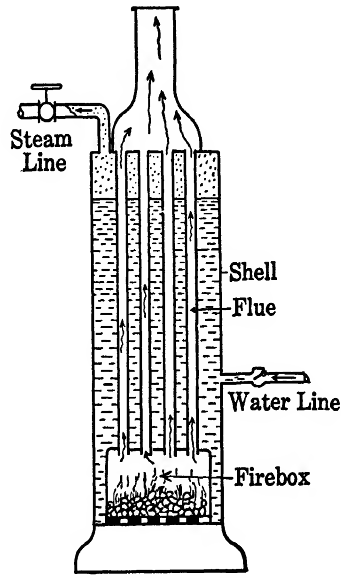


FIG. 2-4. Simple pressure steam boiler.

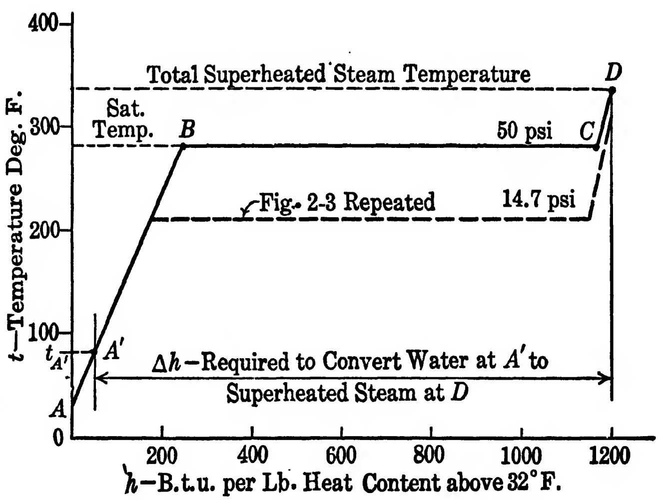


FIG. 2-5. Effect of pressure on t-h relation.

the pipe (engine, heaters, etc.). Figure 2-5 shows the change in condition of the water as it passes through the boiler. Entering at some temperature $t_{A'}$,

it already possesses some heat of the liquid, $h_{A'}$. The heat transferred through the flues is absorbed by the water; first as heat necessary to bring it to saturation at 50 psi. (saturation temperature 281° F), then as heat of vaporization (924 B.t.u. per lb. at 50 psi.) which completes the conversion into steam. As some of the flue is exposed above the water line, the steam conceivably could absorb more heat and become slightly superheated before leaving the boiler. Notice the effect of variation in steam pressure. The line BC is higher in the graph, and shorter, the higher the pressure. This trend continues until, at a pressure of 3206.2 psi., and a temperature of 705.4°, the length $BC = 0$. This is the critical condition where saturated steam and saturated water are the same physically. The locus of points B and C carried upwards to the critical point is often called the *vapor dome.*

2–9. Steam Tables. Unless heated to a high degree of superheat, steam is not governed by the characteristic gas equations. Its thermodynamic properties have been carefully tested on numerous occasions. The results of these tests are available in tables. The properties commonly tabulated are *pressure, temperature, enthalpy, entropy,* and *volume.* It is customary for engineering and industrial use to base these tables on (1) a quantity of one pound of the water substance, (2) a datum of "water at 32° F."

With reference to Figure 2–3, the properties of states such as B and C depend on one variable, i.e., pressure or saturation temperature. Hence, in tables, $t_{\text{sat.}}$ could be the independent variable with p, v, h, and s dependent. In another table of the same data, p might be the argument for determination of $t_{\text{sat.}}$, v, h, and s. As state C represents the addition of a vaporization quantity to state B, volume, enthalpy, and entropy data should each be subdivided into liquid, vaporization, and liquid plus vaporization quantities. The usual tabular nomenclature is to subscript with f for liquid, fg for vaporization, and g for dry saturated steam. Table 2–3 shows the form of a table of saturated steam data with pressure the argument. A similar table with saturation temperature as argument would merely interchange the first two columns, with the data entered being appropriate for uniform temperature increments.

When the state of a vapor is wet and saturated its volume, enthalpy, and entropy are determined by two properties, either pressure and quality or temperature and quality. Generally, wet steam properties are not tabulated since they are so readily computed from the dry steam data.

Example 1: Determine V, h, and s of steam having 90% quality at a pressure of 50 psi. In each case, the wet steam property is the sum of the liquid property plus the proportional part of the vaporization property. Hence:

$$V = v_f + xv_{fg} = .017 + .90 \times 8.49 = 7.66 \text{ cu. ft. per lb.}$$

$$h = h_f + xh_{fg} = 250.1 + .90 \times 924.0 = 1081.7 \text{ B.t.u. per lb.}$$

$$s = s_f + xs_{fg} = .411 + .90 \times 1.247 = 1.533 \text{ B.t.u. per deg. per lb.}$$

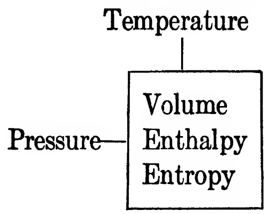
TABLE 2-3. SAMPLE OF THE SATURATED STEAM TABLE

Abs. Press., lb./sq. in.	Temp., Fahr.	Specific Volume			Enthalpy			Entropy		
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
1	101.7	.016	333.6	333.6	69.7	1036.3	1106.0	.132	1.846	1.978
5	162.2	.016	73.50	73.52	130.1	1001.0	1131.1	.235	1.609	1.844
10	193.2	.017	38.40	38.42	161.2	982.1	1143.3	.282	1.504	1.788
14.7	212.0	.017	26.78	26.80	180.0	970.3	1150.3	.312	1.445	1.757
20	228.0	.017	20.07	20.09	196.2	960.1	1156.3	.336	1.396	1.732
30	250.3	.017	13.73	13.75	218.8	945.3	1164.1	.368	1.331	1.699
40	267.2	.017	10.48	10.50	236.0	937.7	1169.7	.392	1.284	1.676
50	281.0	.017	8.49	8.51	250.1	924.0	1174.1	.411	1.247	1.658

TABLE 2-4. SAMPLE OF THE SUPERHEATED STEAM TABLE

Abs. Press., lb./sq. in.		Temperature, degrees Fahrenheit							
		220°	240°	260°	280°	300°	400°	500°	600°
1	v	404.5	416.5	428.4	440.4	452.3	512.0	571.6	631.2
	h	1159.5	1168.5	1177.6	1186.7	1195.8	1241.7	1288.3	1335.7
	s	2.065	2.078	2.091	2.103	2.115	2.172	2.223	2.270
5	v	80.6	83.0	85.4	87.9	90.3	102.3	114.2	126.2
	h	1158.1	1167.3	1176.5	1185.7	1195.0	1241.2	1288.0	1335.4
	s	1.886	1.899	1.912	1.925	1.937	1.994	2.046	2.093
10	v	40.1	41.3	42.6	43.8	45.0	51.0	57.1	63.0
	h	1156.2	1165.7	1175.1	1184.5	1193.9	1240.6	1287.5	1335.1
	s	1.807	1.821	1.834	1.847	1.860	1.917	1.969	2.016
14.7	v	27.2	28.0	28.9	29.7	30.5	34.7	38.8	42.9
	h	1154.4	1164.2	1173.8	1183.3	1192.8	1239.9	1287.1	1334.8
	s	1.762	1.777	1.790	1.803	1.816	1.874	1.926	1.973
20	v		20.5	21.1	21.7	22.4	25.4	28.5	31.5
	h		1162.3	1172.2	1182.0	1191.6	1239.2	1286.6	1334.4
	s		1.741	1.755	1.768	1.781	1.840	1.892	1.939
30	v			14.0	14.4	14.8	16.9	18.9	21.0
	h			1169.1	1179.3	1189.3	1237.9	1285.7	1333.8
	s			1.706	1.720	1.734	1.794	1.846	1.894
40	v				10.7	11.0	12.6	14.2	15.7
	h				1176.5	1186.8	1236.5	1284.8	1333.1
	s				1.686	1.699	1.761	1.814	1.862
50	v					8.8	10.1	11.3	12.5
	h					1184.3	1235.1	1283.9	1332.5
	s					1.672	1.735	1.789	1.837

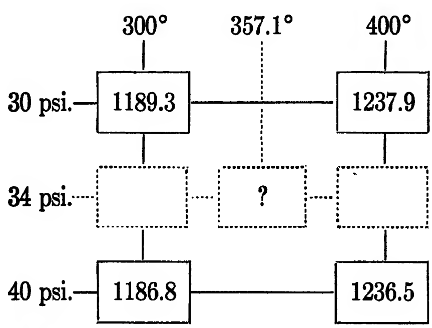
As has been mentioned, the state of superheated steam (similar to *D*, Figure 2.3) is subject to fixation by two independent variables. To exhibit the thermodynamic properties of superheated steam, pressure and total steam temperatures are employed as shown in Table 2–4. For each horizontal pressure row a number of total steam temperatures are given. For each combination of pressure and temperature the table displays the corresponding total volume, enthalpy, and entropy of the superheated steam in this manner.



These v, h, and s data, let it be remembered, represent the total amounts above water at 32° and not the superheat increments. The superheat increment can always be calculated, however, by subtracting the dry saturated steam quantity from the total superheated quantity. Thus, heat of superheat at p_1 and t_1 would be found by subtracting h_g at p_1 from h at p_1 and t_1.

When the data given does not coincide with tabular entries simple proportional ("straight line") interpolation is permissible. In some cases use of superheated steam tables will involve double interpolation, for neither the given pressure nor temperature may correspond to a tabular entry. To illustrate use of the tables two sample examples are here solved.

Example 2: Determine the enthalpy of steam at 34 psi., having 100° superheat. From Table 2–3 it is found by interpolation that the saturation temperature is $250.3 + \frac{4}{10}(267.2 - 250.3)$ or 257.1°. Total steam temperature at 100° superheat is therefore $257.1 + 100 = 357.1°$. Since the desired state (34 psi., 357.1° F) is bracketed by four tabular entries, these will be taken from the table and arranged thus:



Next the enthalpy at 300° and 34 psi. is interpolated, then that at 400° and 34 psi. These are, respectively, 1188.3 and 1237.3 B.t.u. Finally, these two are interpolated for a temperature of 357.1°, giving desired enthalpy of $1188.3 + (57.1/100)(1237.3 - 1188.3)$. This is found to be 1216.2 B.t.u. per lb.

Example 3: How much heat must be transferred to convert water originally at 60° F into wet steam at 50 psi., having a quality of 95%? What is the increase of volume? The heat content of water at 50 psi. and 60° F is practically the same as though the pressure were low enough for 60° to represent saturation (less than 1 psi.). Except where extreme accuracy is required it is usual to assume the enthalpy of sub-cooled water to equal the heat of the fluid, h_f, at the existing temperature. This is readily found in a saturated steam table having temperature the independent variable, but may for ordinary temperatures be assumed as $(t - 32)$ B.t.u. per lb. with little error. Let 2 be the final and 1 the initial state, then

$$\text{Increase of enthalpy} = h_2 - h_1,$$

where $h_2 = h_{f_2} + xh_{fg_2}$ and $h_1 = h_{f_1}$.

$$h_2 - h_1 = 250.1 + .95 \times 924.0 - 28.0 = 1099.9 \text{ B.t.u. per lb.}$$

The increase of volume of the liquid being heated from 60° to saturation at 50 psi. is negligible; therefore, the increase of volume is due only to evaporation. Were the evaporation complete, the increase would be v_{fg}, or 8.49 cu. ft. but, since the steam is only 95% dry, the increase of volume is $.95 \times 8.49 = 8.07$ cu. ft. per lb.

2-10. Steam Calorimeters. Dry saturated *steam* is rarely produced by a boiler. If unequipped with a superheater, the boiler will generate steam which contains droplets of moisture suspended in the steam. Steam exhausted from an engine or turbine has a considerable amount of moisture, the quality often being as low as 85%. It is not possible to judge the quality by inspection of steam, or of pipes in which steam is flowing. A thermometer is of no avail, and yet, there often arises the need for knowing the quality of steam flowing in a pipe line. To meet this need, there have been devised so-called steam calorimeters—devices for experimentally determining the quality of steam. Such calorimeters are made in two forms, each of which has its own particular sphere of usefulness. These instruments, which differ in principle, range of quality measurable, and technique of operation, are known as the *separating* and *throttling* calorimeters.

The separating calorimeter operates on a mechanical principle. It separates the water from the steam by centrifugal force, or difference of density of water and steam, and the principle is simply one of effecting the separation, and then separately measuring the water and dry steam. If A pounds of water are separated from B pounds of dry steam, the quality of the mixture is $\frac{B}{A + B}$. The calculations required in this calorimeter are thus seen to be simple, but the steam must be comparatively wet, because an extremely small amount of moisture cannot be effectively separated from the stream of dry steam. The calorimeter is provided with a catch chamber in which the separated water collects. The amount collected is measured by a graduated gage glass connected to the calorimeter. The dry steam is measured by being passed through an *orifice*. A peculiar property of the orifice is that the weight

of steam passed is proportional to the steam pressure. Using a fixed orifice, a scale of steam flow can be incorporated on the dial of an ordinary steam pressure gage. However, the flow through the orifice is proportional to the calorimeter steam pressure only if the final pressure after passing the orifice is less than 58% of the initial pressure. This is equivalent to a requirement that the steam to be tested be at a pressure of over 10 pounds' gage. For

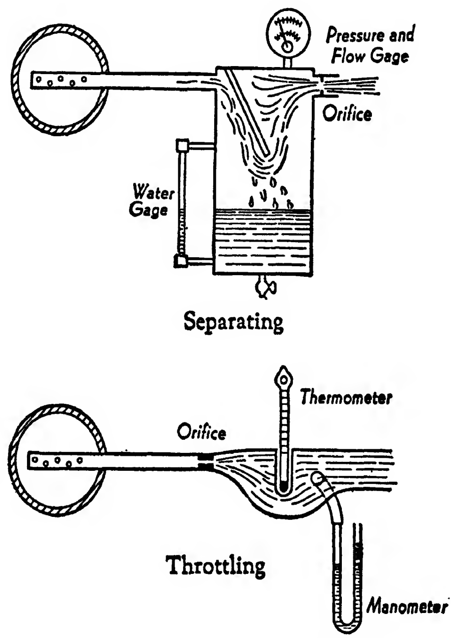


FIG. 2-6. Steam calorimeters.

testing atmospheric exhaust steam, the calorimeter would have to discharge into a region of 12 in. mercury vacuum or better.

The second calorimeter type depends on thermodynamic rather than mechanical action. To *throttle* means to choke. Throttling of a fluid flow occurs when the cross-sectional area of flow is decreased or "choked" by a barrier containing an orifice, or by a partly closed valve, damper, or other gate in the conduit. During a throttling process there is no net change due to mechanical work unless the kinetic energy of the downstream flow is significantly different from that upstream. In processes where steam is throttled significant changes of kinetic energy are virtually non-existent, and so the process may be assumed to occur at constant enthalpy. Pressure, temperature, specific volume, and entropy all change, but enthalpy remains constant.

The throttling calorimeter is not suitable for other than the measurement of steam of very high quality. Its action depends upon the fact that the throttling of high-pressure steam through an orifice to a low pressure leaves

the steam with the same enthalpy at the low pressure that it had at the high. Unless the high-pressure steam exceeds 1800 psi. (a rare case), the heat it contains, if dry, exceeds that of dry saturated steam at atmospheric pressure, so that it may be slightly wet, and yet have enough heat to produce superheated steam after being throttled. Now while the thermometer is of no avail in ascertaining the state of saturated steam, it does, in connection with pressure, definitely determine the state of superheated steam. Unless the steam is initially too wet to superheat the throttled product, the pressure and temperature readings of the low-pressure steam in the calorimeter are sufficient to determine its heat. Taking this, also, as the heat of the high-pressure steam, the quality is computed by the relationship: heat of wet steam = heat of liquid + quality × heat of evaporation. The heat of the liquid and the heat of evaporation are taken from steam tables at the pressure of the pipe line. The accompanying diagrams are intended to be illustrative of the principle, rather than of the actual arrangement of the calorimeter equipment.

Example 1: What is the quality of steam in a pipe line where the gage pressure is 30 psi.? A separating calorimeter extracting a continuous sample of steam from this line separated and collected 2.88 lbs. water during a half hour interval. During this time the calorimeter gage position indicated 4.92. Examination of the scale of the gage dial showed that it was calibrated in pounds flow per 10-minute interval. To apply the simple calorimeter equation it is necessary to refer A and B to the same time interval. Assuming this as 30 minutes, then $A = 2.88$. $B = 3 \times 4.92 = 14.76$; then

$$Quality = \frac{14.76}{2.88 + 14.76} = .837\ (83.7\%).$$

Example 2: A throttling calorimeter being connected to a steam pipe line carrying wet steam at 50 psi. abs. yields the following readings: Calorimeter thermometer 240°, manometer 3.0 in. Hg. What was the quality? Calorimeter absolute pressure was $14.7 + 3.0 \times .491 = 16.2$ psi. Saturation temperature at this pressure (to nearest degree) is 217°. Since the calorimeter steam is actually at 240°, it is seen to be superheated, thus establishing the fact that the pipe line steam quality was high enough to permit use of the throttling calorimeter for its determination. The enthalpy of the superheated steam might be interpolated from tables at 16.2 psi. and 240° total temperature but, as this is sometimes a tedious calculation, an alternative is suggested. Figure 2–3 shows that the enthalpy of superheated steam is the sum of the enthalpy of dry saturated steam and the heat of superheat. The latter quantity is the degree of superheat multiplied by a suitable specific heat. In the use of throttling calorimeters a specific heat of .47 B.t.u. per lb. per degree is justified. Hence enthalpy $h = h_g + .47(240 - 217)$. h_g (from tables) = 1152.2 B.t.u. So, the enthalpy of this throttling process is $1152.2 + .47 \times 23 = 1163.0$. Now, equating this to the high-pressure steam enthalpy,

$$1163.0 = 250.1 + x(924.0),$$

in which x = unknown quality, and 250.1 and 924.0 are, respectively, h_f, h_{fg} at 50 psi. The solution gives $x = 98.8\%$.*

* Any solution yielding $x > 100\%$ implies steam originally superheated.

2–11. Vapor Charts. The properties of vapors set forth in tables are frequently charted, for the graphs, though containing no information not also in tables, allow the reader more readily to visualize the effect of change of one property upon the others. The *t–h* chart has been used in this chapter, but *P–V*, *T–s*, and *h–s* planes provide charts for a greater variety of uses. *P–V* charts are especially useful for positive displacement (engine-type) machines, *T–s* for flow (turbine-type) machines. An area enclosed by a cycle drawn on the *P–V* plane represents foot-pounds of work, that of the *T–s* plane represents work in B.t.u. units if the cycle is reversible.

Probably the most valuable vapor chart is drawn on *h–s* axes since it offers a convenient method of analyzing throttling and constant entropy processes. Generally, the total heat is made the ordinate, and entropy the abscissa. This chart of the properties of vapor is named the Mollier Diagram, and is of considerable use in tracing both theoretical and actual expansions of vapor. A throttled expansion on the Mollier Diagram is parallel to the constant heat lines, and a reversible adiabatic expansion is parallel to the constant entropy lines. Pressure, quality or superheat, and total temperature are shown on the Mollier Diagram as series of lines curved and inclined to the axes. Thus all thermodynamic characteristics of a vapor except volume may be displayed on the Mollier Diagram.

2–12. Refrigerants. A refrigerant is a substance which is suitable as the working medium of a cycle of operations wherein refrigeration is accomplished. To be satisfactory for this service, the refrigerant should be capable of absorbing heat at a low temperature (i.e., the temperatures associated with ice making, cold storage, and other forms of refrigeration) and release it at a higher temperature. This may be accomplished only by a suitable expenditure of energy, in accordance with the second law of thermodynamics, and through processes involving expansion, evaporation, or chemical change. (The last method is not used.) Refrigerants in actual use can be either gases or vapors.

The use of a gas is but little favored in refrigeration, due to the bulk of the equipment necessary in the cycle; however, refrigeration systems using air as a working medium are entirely successful. Most refrigeration equipment operates with a vaporizable liquid as a working medium. The properties that the most suitable refrigerant should possess are:

1. It should be readily obtainable.
2. It should condense at normal cooling water temperatures at relatively low pressures.
3. Its boiling point should be sufficiently low not to require vacuum operation.
4. It should have a high latent heat of vaporization.
5. For the service desired the odor should not be objectionable.
6. It should not seriously interfere with lubrication.

Those refrigerants which are or might be used are as follows:

TABLE 2–5. REFRIGERANTS

Refrigerant	Chemical Formula
Ammonia	NH_3
Carbon Dioxide	CO_2
Sulfur Dioxide	SO_2
Ethyl Chloride	C_2H_5Cl
Methyl Chloride	CH_3Cl
Butane	C_4H_{10}
Propane	C_3H_8
Ethane	C_2H_6
Isobutane	C_4H_{10}
Nitrous Oxide	N_2O
Ether	$(C_2H_5)_2O$
Carbon Disulfide	CS_2
Chloroform	$CHCl_3$
Carbon Tetrachloride	CCl_4
Dichlorodifluoromethane	CF_2Cl_2

PROBLEMS

1. Plot to scale a diagram similar to Figure 2–1 for H_2O. The triple point is at 32° and .0885 psi. Fusion line can be drawn by considering the fusion temperature 32° regardless of pressure (only approximately true). Coordinates of two points on the sublimation line are (1) 0° F, .0185 psi., (2) −40° F, .0019 psi. Other data will be found in text of this chapter.

2. How many heat units are absorbed during the heating of one gallon of water from 60° to 110° F?

3. Ten cubic feet of nitrogen are cooled from 350° F to 120° F at constant volume. Final pressure is 20 psi. gage. How many pounds of nitrogen are in the tank? How much heat was removed during cooling?

4. A liquid in cooling from 175° to 88° F released heat that was measured by absorbing it in water. While 25 lbs. of the liquid was cooled, 67 lbs. of water absorbed the heat with a rise in temperature of from 62° to 90° F. Calculate the specific heat of the fluid.

5. What is the weight of a block of concrete 32 in. x 16 in. x 72 in.? How much heat is absorbed by it under a temperature change of +35°?

6. An iron rod 12 ft. long and 1 in. in diameter absorbs 250 B.t.u. Find its increase in length, to the nearest hundredth of an inch.

7. A mercury barometer reads 29.62 in., while the thermometer attached to it shows the temperature of the mercury to be 70° F. What is the barometer reading corrected to a 32° F mercury column? A 60° F column?

8. A carload of fuel oil (petroleum) is volumetrically measured and found to contain 8022 gallons of oil at a temperature of 40° F. How many gallons would there be if measured at 60° F?

9. A block of ice at 20° F measuring 20 in. x 10 in. x 15 in. is melted to water at 32°. The heat of fusion is 144 B.t.u. per lb. ice. How many B.t.u. were absorbed by the H_2O during this change?

10. An ice plant has a capacity for making 2 tons of ice per 24 hours with the equipment working steadily at extracting heat energy from the H_2O in the cans. Water is first put in the cans at 60° F, then chilled, frozen, and sub-cooled to 25° F. Heat of fusion 144 B.t.u. per lb. How many B.t.u. are removed by the process per 24 hours? Per minute? How many B.t.u. per minute for each ton per day capacity? (Often refrigeration literature cites 200 B.t.u. per minute as equivalent to a *ton* of refrigeration.)

11. Air used in a hot air heating system is considered to remain at constant pressure. How many pounds of air per hour must be continuously circulated in order to replenish heat losses of 180,000 B.t.u. per hour from the building? Air is heated from 65° F to 150° F by the furnace. What is the volume of the 150° air, cu. ft. per minute?

12. Multiply the constant R by the molecular weight for each of the gases in Table 2–1. Comment on the results.

13. What volume does a mol of gas occupy at 60° F and 120 psi.?

14. How many mols are there in 2 cu. ft. of oxygen at 1200 psi. and 60° F?

15. Find the weight of methane passing through a meter per hour. The gas is at 15.5 psi. and 60° F, and the meter indicates a flow of 7.5 cu. ft. per minute.

16. One pound of air undergoes a constant pressure change at atmospheric pressure, rising in temperature from 60° F to 150° F.

a. What is the initial specific volume?
b. What is the change in volume in cu. ft.?
c. How many foot-pounds of expansion work are done?
d. How much heat is added?

17. A pound of oxygen is heated at constant volume from 60° F to 300° F. Find Δh and Δs.

18. Ten cubic feet of CO_2 expand adiabatically from 120 psi. 120° to 14.7 psi. Explain why $\Delta Q = 0$ and $\Delta s = 0$. What is the final volume? Use the polytropic equation $PV^\gamma = C$. Using the general gas law twice, find the weight of CO_2 and the final temperature.

19. Helium fills a tank 3 ft. in diameter by 4 ft. long at 120 psi. Temperature 60° F. It is to be heated until the pressure doubles. How much heat is added? Find Δh and Δs.

20. Find the density before and after heating a pound of air from 65° to 95°. Also find Δu, Δh, Δs, assuming that the change took place at constant pressure of 14.7 psi.

21. From tables obtain and record the following for a pressure of 30 psi.: (a) heat of the liquid, (b) entropy of vaporization, (c) enthalpy of dry and saturated steam.

22. Consult steam tables for the following characteristics for a pressure of 45 psi.: (a) boiling temperature, (b) heat of vaporization, (c) entropy of vaporization.

23. Compute the change of entropy, when saturated liquid is converted into dry saturated steam, by the equation $\Delta s = \Delta Q/T$, in which ΔQ will be h_{fg}. Do this for (a) 20 psi., (b) 45 psi., then check with tabular values for s_{fg}.

24. Draw t–h graph representing the heating of a pound of water, originally at 32° F, until it becomes dry saturated steam at 40 psi. 1 in. = 100°, 1 in. = 200 B.t.u. Label fully.

25. Draw *t–h* graph representing the conversion of one pound of water, originally at 32° F, into superheated steam at 50 psi. and 200° superheat. 1 in. = 100°, 1 in. = 200 B.t.u.

26. On the *T–s* plane construct the "steam vapor dome" by plotting s_f and s_g against $t_{sat.}$ 1 in. = 200°, 1 in. = 0.3 B.t.u./degree.

27. Draw *t–h* graph representing the conversion of one pound of water originally at 120° into wet steam of 95% quality at 200 psi. 1 in. = 100°, 1 in. = 200 B.t.u.

28. Draw *t–h* graph representing the conversion of one pound of water originally ready to boil at 14.7 psi. into superheated steam at 14.7 psi. and 500° F. 1 in. = 100°, 1 in. = 200 B.t.u.

29. The boiler shown in Figure 2.4 is to operate at a pressure of 140 psi. gage. Feed water enters it at 60° F. What is the increase of enthalpy per pound passing through the boiler if the outgoing steam has a quality of 95%?

30. A kettle (Figure 2.2) is charged with one gallon water at 70° F. How many heat units will be absorbed before the contents are converted to dry saturated steam?

31. Find the entropy of steam at 400 psi. gage, 200° superheat.

32. Find the volume of steam at 20 psi. gage, 200° superheat.

33. A superheater converts boiler steam of 96% quality into superheated steam of 600° F total temperature, all at 500 psi. abs. How much heat is added per pound?

34. What volume is occupied by 10 lb. of wet steam having a quality of 85% at a pressure of 5 psi. abs.?

35. Dry saturated steam at 40 psi. is changed by expansion to 14.7 psi. If the entropy is the same after expansion as before, what is the quality after expansion? The volume per pound? The temperature?

36. Diagram (no scale) a portion of the Mollier Chart for steam and show on it a process in which (a) pressure is increased at constant enthalpy, (b) pressure is decreased at constant entropy, (c) enthalpy is increased at constant temperature. Show the *direction* of the process by an arrowhead.

37. Sketch part of the Mollier Chart (no scale) so as to show a process in which (a) enthalpy is increased at constant entropy, (b) flow is throttled at constant enthalpy, (c) temperature is increased at constant pressure. Show the *direction* of the process by an arrowhead.

38. Steam of unknown quality is blown through a separating calorimeter which collected 2.21 lbs. water in 25 minutes while exhausting dry steam at the rate of 12 lbs. per half hour. What was the quality?

39. Steam of 85% quality *enters* a separating calorimeter at the rate of 32 lbs. per hour. The pressure gage is calibrated in terms of pounds dry steam per 10 minutes. Diagram the separating calorimeter and label it to show (1) quantity of water which should be collected in 20 minutes, (b) position of pointer on flow gage.

40. Draw diagram of a type of throttling calorimeter that has a discharge outlet so unrestricted that atmospheric pressure would exist within the calorimeter body. Show the reading the thermometer should have if the pipe line contained steam of 98% quality at 50 psi.

41. A throttling calorimeter is used to test the steam on a line where the pressure is 100 psi. gage. Calorimeter pressure is 2.5 in. Hg gage, temperature 238°. What was the quality of the pipe line steam?

42. Steam is throttled from 45 psi. to the atmosphere by a partially opened valve. The temperature of the throttled steam is 212°. Should a separating or throttling calorimeter be used to measure the quality of the 45 psi. steam? Why?

43. Steam flowing past an engine governor is cut from 250 psi. to 150 psi. It was originally 98% dry. Considering this to be a throttling process, what is the state of the steam leaving the governor and entering the engine? Supply a labeled section of the Mollier Chart to illustrate what happened to the steam.

44. Dry saturated steam expands from 50 psi. to 14.7 psi. at constant entropy. (a) Obtain the initial entropy from tables, set it equal to the final entropy, and calculate the final quality. (b) With the final quality determine the enthalpy and the change of enthalpy during the process.

45. A pound of superheated steam at 100 psi. and 600° F is cooled at *constant volume* until it is exactly dry and saturated. At what pressure does it become dry and saturated?

46. Two pounds of ice at 32° (heat of fusion 144 B.t.u. per lb.) are put into 15 lbs. of water at 60°. What is the final condition of thermal equilibrium, i.e., temperature, water present, and ice present? Solution will rely on the *First Law of Thermodynamics*, as will those of succeeding problems.

47. A pound of ice at 32° (heat of fusion 144 B.t.u. per lb.) is put into 5 lbs. of water at 40°. What is the final condition of thermal equilibrium, i.e., temperature, water present, and ice present?

48. Fifty pounds of water at 60° F are sprayed into an open chamber containing a pound of steam at 14.7 psi. and 90% quality. What is the final condition of thermal equilibrium?

49. After 75 lbs. of water have condensed a pound of steam (at atmospheric pressure) the temperature of the mixture came to 102° F. The water was originally at 85° F. What was the state of the steam (i.e., what quality or superheat)?

50. Drinking water is to be chilled, at the rate of a gallon a minute, from 80° F to 40° F. How much refrigeration capacity (counting 200 B.t.u. per minute = one ton of refrigerating capacity) is required? How much ice (at 32°) would be used for the same purpose if the ice were to be added to the water in a chilling tank.

CHAPTER 3

Fuels and Combustion

3–1. Fuel. Chemical reactions are often accompanied by manifestations of energy. This phenomenon underlies a special field of science—thermochemistry. Many reactions are accompanied by an evolution of heat. These are termed *exothermal* reactions. During others heat is absorbed by the reaction, these being *endothermal* in nature. One phase of thermochemistry is vital to heat power study since it is the common method of originating dynamic energy. This is the oxidation of a fuel for the purpose of generating heat energy by the violently exothermal nature of the reaction. A fuel may be defined as a substance which in rapid chemical union with oxygen produces heat energy in useful quantity and at an elevated temperature. Several elements have a sufficiently exothermal reaction with oxygen, but the two elements which compose most commercial fuels are carbon and hydrogen. These exist as fuels in their pure form, but are more frequently found either combined with each other (as in CH_4, C_2H_6) or with other non-fuel elements (CO, C_2H_2OH).

Fuels may be grouped variously; the following tabulation is illustrative of the most common classification systems.

A. Physical state.
  1. Solid. Coal, wood, coke, etc.
  2. Liquid. Petroleum, alcohol.
  3. Gaseous. Illuminating gas, natural gas.

B. Origin.
  1. Natural. Coal, petroleum, wood.
  2. Artificial. Principal products such as alcohol, illuminating gas, acetylene, gasoline. By-products such as blast furnace gas, refinery residue, sawdust.

Coal and petroleum, or products derived from them, are the principal fuels of the present time. Natural gas, wood, and alcohol, although common fuels, are of lesser importance.

3–2. Coal. Coal is a combustible substance of organic origin which occurs as beds or "seams" and which has a variable physical and chemical composition, including small amounts of mineral or non-combustible matter. Although the combustible constituents of coal are of organic origin, the geological

processes involved in its formation act so slowly and over such a long period of time that coal is, economically speaking, non-reproducible.

It is known that the origin of coal is to be sought in the vast and luxuriant vegetation which flourished over portions of the earth's surface in past geological ages, for the imprint of leaves of giant tree ferns of a form similar to those now found only in tropical jungles can often be seen on the face of blocks of coal at the mine. Undoubtedly the somewhat higher mean temperature of the earth, coupled with the prevalence of a more steamy atmosphere, produced conditions which led to very rapid growth of vegetation. Such conditions are approximated only in hot-houses, so it is probable that the coal-forming epochs are definitely of the past, and that such coals as are used constitute a depletion of a fixed natural resource. Through the centuries of the carboniferous age there was produced a rapidly accumulating mass of decaying vegetation, a great deal of which was undoubtedly also preserved from complete decay by having air excluded from it either by its being partially submerged or being covered with a considerable thickness of superimposed matter. The process of decay soon converted the cellulose, lignin, and proteins of the original vegetable debris into peat bogs which, due to some subsequent earth movements, were either rapidly or gradually covered with inorganic material which, in time, produced sedimentary rocks such as shale or sandstone. Coal is found today in bedded deposits, or seams, in company with shale, sand-stone, limestone, and other sedimentary products. The pressure of the overlying strata, together with some possible small amount of residual decay, created an increase of temperature in the peat which slowly accomplished its conversion into coal. It is entirely conceivable that the variations in overlying strata, in condition of the bed when overlaid, and in subsequent rise of temperature, are accountable for the differences in coal as it is brought from the mines today. During the process of this chemical change, percolation of water through the incipient coal seams probably carried into the coal many of the minerals which, in the aggregate, produce the ash which is common to all coals.

The classification of coal is primarily in terms of its origin, both as to (1) composition, or the original types of vegetable material composing it, and (2) the degree of *metamorphism* which it has undergone since burial. A simple but practical classification of coal according to its origin and chemical composition is as follows.

1. Peat. Partially carbonized vegetable matter, such as accumulates in a bog.

2. Lignite. Brown coal and lignite are formed by the burial and consequent compression of peat by overlying sediments (formations), the peat substances being thus compacted and changed, first to brown coal or *lignite*, and then to black lignite or sub-bituminous coal. Although lignites are

thus the parent forms of the higher grades of coals, even pure lignite is a relatively poor form of fuel because of its high water content and consequent low calorific value.

3. Bituminous coal. This is "soft coal" to most persons. It is found in different gradations (ranks), and is generally characterized by dull black appearance and easily fractured (but not crumbling) structure. It is widely distributed throughout the world, and is the principal type of coal used for the production of steam. Bituminous coal contains 70 to 80% of fixed carbon, and 30 to 20% of volatile hydrocarbons. When most bituminous coals are heated in a closed retort this volatile matter is driven off as gas and *coal tar.* The residue of fixed carbon may form a coherent porous mass called *coke.*

4. Anthracite (Hard Coal). When bituminous coals have been heated and compressed (by natural geological causes) so that they have changed from the finely joined "soft coals" to the massive and hard form, they are called anthracite, which is relatively hard, clean and moisture free, low in volatile matter and high in fixed carbon. Anthracite grades to graphite which is high in fixed carbon, combustible only at high temperatures, and classed as a refractory.

Coal contains *carbon, hydrogen, oxygen, nitrogen, sulfur,* and various mineral substances such as silica, alumina, and iron (ferric) oxide. All of the mineral substances are included in the descriptive term *ash.* Hydrogen is present in coal in two forms, called *free* and *combined,* although actually it is in two methods of combination. Hydrogen is combined with oxygen forming the moisture in coal, and all coal contains more or less moisture. Hydrogen is also combined with carbon, forming the hydrocarbon, or volatile portion of coal. A very important difference, however, exists in these two combinations of hydrogen. In one, water, it is completely oxidized, and must be considered incombustible; in the other, the hydrocarbons, the hydrogen is combustible. Occurrence of ash in coal is more or less accidental. Coal rarely contains more than 4% of sulfur or 2% of nitrogen. Only a small portion of the carbon content of coal is accounted for in the volatile portion (the hydrocarbons); in fact, the largest portion of the heating value of coal is obtained from the free or "fixed" carbon. Ash and moisture-free fuel is designated "combustible." A sample analysis of anthracite coal is: 2% moisture, 5½% volatile material, 86½% fixed carbon, 6% ash. Typical semi-bituminous coal, the principal steaming coal, would be covered by the following analysis: 3% moisture, 18% volatile material, 75% fixed carbon, 4% ash.

The commercial production of coal is a vast and highly intricate business. The seams which are worked are usually less than ten feet thick, and are generally overlaid with a rock cap which is shored up as the coal is removed. Some coal is mined by the open or strip method, in which the overlying earth is stripped off by power shovel. Coal is undercut or blasted out of the face

of a seam and loaded onto cars which convey it to a plant where the slate is removed before the coal is broken and graded.

Bituminous coals are produced in the following sizes:

Run of Mine. This coal is as mined, unscreened, and varying in size from large lumps to slack.

Lump. Coal which passes over a $1\frac{1}{4}$-in. screen.

Nut. Cut coal of a size which passes through a $1\frac{1}{4}$-in. screen, but is retained on a $\frac{3}{4}$-in. screen.

Slack. All coal passing the $\frac{3}{4}$-in. screen.

Anthracite coal, in addition, is produced in the finely graded sizes called Pea and Buckwheat.

Coal is the source of a great many important and useful by-products which are derived either from distillation or carbonization of bituminous coal. Amongst the more important of these might be mentioned the following: illuminating and fuel gas, coke, ammonia, analine and other dye stuffs, explosives, pitch, cresoline and paint compounds, antiseptics, tar.

3-3. Fuel Oil. Petroleum, like coal, is thought to have originated from buried organic matter. However, theories of the character of this substance and its synthesis into petroleum oil are not as well formulated as for coal. Petroleum is a mixture of hydrocarbons held in porous subterranean formations from which it is reclaimed by the drilling of wells into the oil-bearing structure. As with coal, the analyses of oils from different sites may vary somewhat.

The principal production fields in the United States are in Pennsylvania, Ohio, Texas, and California. Petroleum is composed of a series of hydrocarbons in various proportions. A typical analysis of crude oil would be 84% carbon, 13% hydrogen, 1% sulfur, 1% nitrogen, 1% oxygen. Practically all of the hydrogen is "free" (that is, it is in a state of combination with carbon, in which most or all of its heat of combustion is available), and since it is in larger proportions by weight than in coal, fuel oil might be expected to have a considerably higher heating value than coal. The principal uses of fuel oil at the present time are for Diesel engines, for firing in the furnaces of steam boilers, and for use in domestic oil-burning furnaces.

The petroleum oil obtained from wells is rarely suitable as a fuel oil in its crude form. It can be refined by fractional distillation so as to remove some of the heavier objectionable compounds. Fuel oils are crude oils without the heavier hydrocarbons originally in petroleum. Also, some good fuel oil comes out of the "cracking" and "hydrogenation" processes used to produce gasoline.

The value of a fuel oil may be gaged by the following properties.

1. Calorific value. This, of course, is important because it shows the amount of energy available. The calorific value of fuel oils is from 18,000 to 20,000 B.t.u. per lb.

2. Viscosity. Viscosity is one of the most important qualities of a fuel oil. It determines the fluidity of the oil and is a fair indication of how readily the oil will atomize. A very viscous oil may prove troublesome to handle without heating to a point where viscosity is more satisfactory. Viscosity is generally stated in terms of seconds, Saybolt Universal, at some specified temperature, usually 100° F.

3. Ignition quality. This characteristic measures the ability of fuel oil to ignite spontaneously in an engine cylinder. Good ignition quality is represented by high *cetane number*. Ignition quality is an index to possible detonation, ease of starting, and smoky exhaust.

4. Carbon residue. The standard test for this characteristic is the Conradson Test which gives an indication of the amount of carbon that may be deposited as a scale in an engine combustion chamber. Although low carbon residue is a desirable quality and will yield better engine operation, the lower cost of less completely distilled fuels makes them more economical in use.

5. Solid impurities: Ash and asphalt content. The effect of these is to cause deposits in combustion chambers, around the piston rings, and in burner tips, necessitating increased maintenance.

6. Water content. Few oils are free from water, but a good oil will not contain more than one per cent of water. Besides forming sludge, the water content decreases the calorific value of a fuel.

7. Sulfur and acid. These components will cause corrosion under certain conditions.

8. Gravity. The Baumé scale is one of specific gravity.

$$\text{Degrees Baumé} = \frac{140}{\text{Specific gravity}} - 130$$

(For liquids lighter than water).

The specific gravity is referred to water as 1. Suitable fuel oils range from 15° to 43° Baumé. The Baumé reading is useful in determining whether the oil is burdened with an undue amount of heavy residue which is difficult to burn successfully.

9. Flash and boiling points. These indicate the degree to which a fuel can be vaporized; also the inflammability and hence the fire risk of storage and handling.

3–4. Gasoline. Gasoline, as one of the products of crude oil, is required today in such quantities for motor vehicles that were the crude oil to be separated into its commercial components (such as petroleum ether, gasoline, kerosene, etc.) by *distillation* or *fractionation* through heating and condensation, there would be an uneconomic excess of the products other than gasoline. Petroleum, the natural crude oil, is a complex mixture of hydrocarbons, which

differ in composition and structure. When petroleum is distilled, the more volatile fractions, such as gasoline, are found to be composed, in general, of hydrocarbons of simpler molecular structure—that is, those having molecules which contain a comparatively small number of atoms. In general, the hydrocarbon molecules having larger numbers of atoms are more largely concentrated in the less volatile petroleum products. In the cracking process for making gasoline these larger molecules are broken up to produce, in part, a greater yield of gasoline and other low-boiling fractions. One of the simplest methods of cracking is by heating under pressure. By efficient cracking gasoline yields of 60% or more have been obtained from certain crudes.

Gasoline is obtained from four main sources: (1) natural gas or casinghead gasoline, by condensation under pressure or by adsorption of the liquefiable constituents of natural gas, (2) straight-run gasoline, by the fractional distillation of crude petroleum, (3) cracked gasoline by (a) thermal (heat-pressure) or (b) catalytic decomposition of crude petroleum or its distillate fractions, (4) synthetic gasoline by such reactions as (a) polymerization of hydrocarbons of lower, to those of higher, molecular weight, (b) hydrogenation of producer gas (Fischer-Tropsch), of petroleum or coaltar distillate fractions, or of coal (Bergius). Cracked and synthetic gasolines show distinct and remarkable increases in production with resulting economy in the utilization of crude petroleum.

Catalytic cracking is conducted in the vapor phase, and therefore the distillates used must be those that can be vaporized at the operating temperature of the process.

Specifications for various types of gasoline more or less commonly used in commerce involve color, odor, gums, distillation range, doctor test (for certain sulfur compounds), corrosion (of copper) test, acidity, and sulfur (total). Of these, the most important is the distillation test, which is a measure of the volatility and the range of volatility of the gasoline. Various grades, based upon different distillation ranges, are recognized. The necessity for such specifications is demanded by the extensive use of gasoline in internal combustion engines for motive power in automobiles, trucks, and airplanes. Since gasoline is a mixture of different compounds, no one chemical formula may be applied to it. When it is desirable to represent it as a single compound it is usually assumed to be C_8H_{18}.

3–5. Gaseous Fuels. There are many gases and mixtures of gases that contain sufficient carbon and hydrogen to be considered fuels. Some of these are of natural origin and as such are often almost pure fuel. By-product gas is usually so diluted with nitrogen or products of combustion that its economical employment is limited to the industry in which it originated. Gases like hydrogen, acetylene, or propane are specially manufactured in the pure state and compressed into steel tanks at very high pressure. The purchaser con-

nects his gas-using equipment (stove, welding torch, etc.) to the tank through a pressure-regulating valve which maintains a steady low pressure on the discharge from the tank.

Natural gas is usually found in geological formations which also contain oil. A cross-section of the typical dome formation is shown herewith.

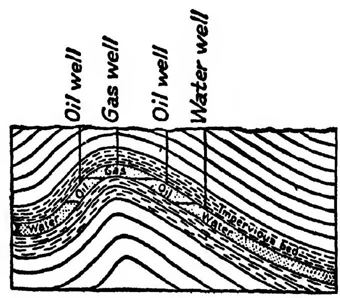


FIG. 3-1. Structure section to show a typical occurrence of gas, oil, and water in an anticlinal structure. (After U. S. Geological Survey.)

3-6. Gas Producer. The gas producer is used to manufacture *fuel* gas from coal, steam, and air. It consists of a vertical cylinder which is lined with fire brick and charged with coal. Coal is fed at the top of the producer, and steam and air are injected at the bottom. The steam and air injected from below pass up through the combustion region, during which the oxygen combines with the carbon of the burning coke bed, forming carbon dioxide, and the steam becomes very highly superheated. When these gases pass upward through the middle zone, the incandescent coke there reduces the carbon dioxide to carbon monoxide, thus

$$CO_2 + C = 2CO$$

and the steam to hydrogen and carbon monoxide, thus

$$H_2O + C = H_2 + CO.$$

The composition of the gas obtained from a producer depends to a great extent on the control of these reactions by variation in the relative proportions of steam and air. A typical analysis of such a gas is: *carbon monoxide* 25%, *methane* 2%, *hydrogen* 15%, *carbon dioxide* 8%, *nitrogen* 50%. The large nitrogen content is due, of course, to the use of air for combustion in the producer. With so large a portion of the analysis contained in this inert gas, it is not surprising that the heating value of producer gas is low when compared to gases more completely composed of combustible elements. The average heating value of producer gas is 150 B.t.u. per cu. ft.

3-7. Fuel Analysis. One of the methods of reporting the analysis of a fuel is on the basis of the chemical elements present and their proportions by weight. Since such an analysis reports the composition of a substance in terms of its ultimate elements, it has been called the *ultimate analysis*. Essentially, it is a chemical analysis. The analysis of a fuel, solid, liquid, or gaseous, as the case may be, for the purpose of resolving it into an ultimate analysis, is a process requiring the trained knowledge of the chemist, the apparatus of a well-equipped chemical laboratory, and no inconsiderable perfection of tech-

TABLE 3–1. FUEL ANALYSES

Percentages as given are by weight.

I. Pennsylvania Anthracite.

Proximate		*Ultimate*	
Fixed carbon	86.24	Carbon	86.37
Volatile matter	5.67	Hydrogen	2.70
Moisture	2.19	Oxygen	3.55
Ash	5.90	Nitrogen	0.91
		Sulfur	0.57
Total	100.00	Ash	5.90
		Total	100.00

II. West Virginia Semi-bituminous (Pocahontas Field).

Proximate		*Ultimate*	
Fixed carbon	73.02	Carbon	80.70
Volatile matter	16.36	Hydrogen	4.20
Moisture	2.18	Oxygen	4.53
Ash	8.44	Nitrogen	1.44
		Sulfur	0.69
Total	100.00	Ash	8.44
		Total	100.00

III. Indiana Bituminous.

Proximate		*Ultimate*	
Fixed carbon	41.04	Carbon	62.36
Volatile matter	38.59	Hydrogen	5.39
Moisture	9.69	Oxygen	15.50
Ash	10.68	Nitrogen	1.28
		Sulfur	4.79
Total	100.00	Ash	10.68
		Total	100.00

IV. Certain Hydrocarbon Compounds.

Ultimate

Octane (approximates gasoline)	
Carbon	84.2
Hydrogen	15.8
Total	100.0
Methane (marsh gas)	
Carbon	75.0
Hydrogen	25.0
Total	100.0
Butane (fuel gas)	
Carbon	82.8
Hydrogen	17.2
Total	100.0

V. Pennsylvania Crude Oil.

Ultimate

Carbon	84.90
Hydrogen	13.70
Oxygen	1.40
Sulfur	0.00
Total	100.00

nique. For this reason, the taking of an ultimate analysis has become a specialized task. Through several years of fuel research and experimentation, many data on the ultimate analyses of coals from different seams have been accumulated, and it is safe to say that very few commercial seams of the present time are without a published analysis of their typical product.

Combustion calculations are essentially calculations of chemical reactions. Any quantitative work starting with chemical reactions must rest on a knowledge of weights of the elements entering into the reaction. It is to be expected, therefore, that the ultimate analysis would be required for combustion calculations. The *proximate analysis* of coal is the determination of its content of moisture, volatile material, fixed carbon, and ash. Much information regarding the firing characteristics and fuel value of coal will be conveyed by the proximate analysis. The test for proximate analysis may be obtained with a small amount of laboratory equipment. For that reason it is useful for checking, from time to time, the quality of coal bought. Typical proximate and ultimate analyses of some common fuels are shown in Table 3–1.

3–8. Heating Value. The heating, or calorific, value of a fuel is the quantity of heat produced by the combustion, under specified conditions, of unit weight or volume of the fuel. The heating value of a fuel may be calculated by formula which may be derived for any fuel by multiplying the fractional part of each chemical element present by its heating value per unit weight, and adding the products for all combustible elements in the fuel. Thus for coal, whose combustible elements consist of carbon, hydrogen, and sulfur, the heating value in B.t.u. per pound of coal is:

$$\text{H.V.} = 14{,}540\text{C} + 62{,}000\,\text{H} + 4{,}000\text{S}.$$

The numbers in the above formula are the heating values per pound respectively, of carbon, hydrogen, and sulfur. In the use of this formula, it is essential that only that portion of the element that is actually free to burn be employed. For example, all coal contains some moisture. Now the hydrogen present in this water is not free to burn (i.e., it is already combined with oxygen). Therefore the figure used for H in the foregoing formula should not include the hydrogen present as water. The correction is made by deducting one-eighth of the oxygen content from the total hydrogen. The moisture which contains the combined hydrogen is one-ninth hydrogen and eight-ninths oxygen.

Heating value by formula will not necessarily be the same as that obtained experimentally with the fuel calorimeter. The difference lies not in the accuracy of the experiment, nor of the calculation, but in the possible endothermic or exothermic reactions which take place when a compound fuel, such as a hydrocarbon, is burned. The volatile matter of coal must be broken down into the elements of carbon and hydrogen by heat-absorbing action

before they may reunite with the oxygen during combustion. For this reason, experimentally determined heating values are less than those which are computed by a formula which makes no reference to endothermic or exothermic reactions. Approximate heating values of some of the common fuels are: coal, 13,000 B.t.u. per pound; natural gas, 1000 B.t.u. per cubic foot; artificial gas, 300 B.t.u. per cubic foot; gasoline, 20,000 B.t.u. per pound; wood, 5000 B.t.u. per pound.

Many thermodynamic analyses require the use of a "lower heating value" which may be obtained from the above values by subtracting an allowance for the latent heat of evaporation of any steam in the products of combustion produced by the combustion of hydrogen. Lower heating value might be thought of as "sensible heating value."

Example: The higher and lower heating values of coal No. II, Table 3–1, will be computed.

$$\text{Higher H.V.} = 14{,}540 \times .8070 + 62{,}000\left(.0420 - \frac{.0453}{8}\right) + 4000 \times .0069$$
$$= 14{,}000 \text{ B.t.u. per lb.}$$

To compute the Lower H.V. deduct for steam in the products of combustion at the rate of 1060 * B.t.u. per lb. Since we can expect 9H lb. H_2O per lb. coal burned,

$$\text{Lower H.V.} = 14{,}000 - 9 \times .0420 \times 1060 = 13{,}600 \text{ B.t.u. per lb.}$$

The heating value of a *fuel*, in terms of heat units such as the British Thermal Unit, may be obtained experimentally with the use of a fuel *calorimeter*. Fuels exist in solid, liquid, and gaseous states. The conditions of measurement of a gas call for an instrument different in many essential respects from that which would be satisfactory for a solid or liquid fuel, principally because it is impractical to measure out a gas in definite isolated quantities, as is so easily done with liquids and solids. Hence a gas calorimeter is a continuous flow instrument, whereas the liquid and solid fuel calorimeter is an intermittent type, wherein a known weight is burned. Nevertheless, the principle underlying the measurement of heat in all fuel calorimeters is the absorption of heat by water, creating a temperature rise. Measurements of the quantity of water and temperature rise are used directly to determine the heat units, since a unit of heat raises the temperature of a unit weight of water one degree on the average.

The Junker's gas calorimeter is a chamber wherein a known metered flow of gas is burned in an efficient type burner, with liberation of heat which is absorbed in water as the products of combustion flow through the tubular passages of the calorimeter. The water which absorbs the heat is likewise flowing steadily through the calorimeter, being separated from the gas by

* An assumption for h_{fg} under actual furnace conditions.

the walls of the tubes. The temperature of the water entering and leaving is measured by thermometers, while the rate of flow of water is measured by catching and weighing it. Thus the heat measured is a rate of heat liberation obtained by multiplying the rate of flow of water per minute by the temperature rise in degrees. This quantity, when corrected for radiation, moisture condensation, emergent stem, and other conditions, becomes the heating value of the gas.

The intermittent type fuel calorimeter consists of a bomb which is charged with fuel and oxygen, and immersed in a bucket of water, which serves to absorb the heat when the charge inside the bomb is ignited. The temperature rise of the water is observed and the heat release determined by multiplying that rise by the known weight of water. That heat, divided by the weight of the measured sample of fuel, is the heating value—in the rough. Several corrections must be applied.

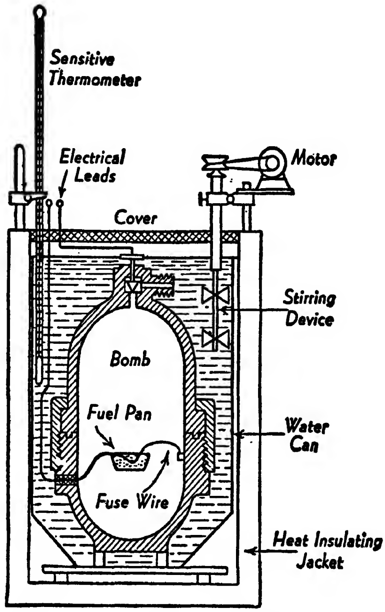


FIG. 3–2. Emerson bomb calorimeter.

The figure illustrates a bomb calorimeter. A carefully weighed sample of the fuel to be tested is placed in the pan within the steel bomb. A length of calibrated fuse wire is looped through the fuel in the pan, then the bomb is screwed tightly together. It is next connected to a tank of oxygen and charged with that gas to a pressure of several atmospheres, after which it is removed and immersed in the water of the calorimeter. When the water and calorimeter have arrived at room temperature, the electric leads are connected to a source of voltage through a switch, a stirring device is started, and the circuit is closed. The rush of current heats the fuse wire, raising it to a temperature sufficient to ignite the fuel in the presence of oxygen. The fuel then burns rapidly—almost explosively—and the heat it liberates is absorbed by the water. Temperature rise is measured by a sensitive thermometer. A large amount of water is used, so that the temperature rise will be very small. Corrections must be made for radiation, emergent thermometer stem, heating value of the fuse wire, water equivalent of the calorimeter, and energy input of the stirring device.

3–9. Combustion. As has been mentioned, a fuel is a substance capable of considerable exothermal heat release upon oxidation. The action, called combustion, might therefore be defined as the rapid chemical union of a fuel

with oxygen, accompanied by the release of substantial quantities of heat energy at high temperatures. The fuels and oxygen molecules do not spontaneously unite until the molecular speeds are high enough (i.e., at high temperatures) that the molecular impacts break down the molecular structure of the separate elements. A new molecular arrangement then appears as the product of combustion. For example, from H_2 and O_2 upon combustion H_2O appears; from C and O_2 combustion produces CO_2. The potential molecular energy of the product of combustion molecules is much less than that of the original elements. The difference was evolved as heat of combustion.

The temperature necessary to start the reaction is the ignition temperature. Ignition is the initiation of combustion. It is necessary for the inception of any flame and is accomplished by raising a mechanical mixture of the combining substances, such as carbon and oxygen, to the "ignition temperature." This temperature may be defined as the lowest temperature which will cause flame to start and spread through a combustible mixture. The ignition temperature varies with the substance. Some typical values in open air are: *

	°F
Carbon	750
Carbon monoxide	1250
Hydrogen	1090
Sulfur	470
Isoöctane	980
Heptane	500
Phosphorus, yellow	93

The most common source of high temperature energy used for exciting molecular activity to the point of ignition is the ordinary match. On account of its low ignition temperature, yellow phosphorus was at one time employed as the igniting element of matches; however, its use was hazardous, and in modern match manufacture the tip is composed of a mixture of various materials, including phosphorus, glue, ground glass, chlorate, and potash. The common match is composed of a parafined splint dipped into a combustible compound which, upon hardening, forms the bulb of the match head. The head is redipped, to form an eye, in a compound which will ignite spontaneously by friction against a rough surface. In safety match manufacture, the phosphorus compound is usually affixed to the side of the box and the head contains the oxidizing agent.

Rays of the sun, properly focused through a lens, are capable of heating some substances to their ignition points. Mechanical (flint and steel) and electrical (spark) energy can be used for the same purpose, as can also compression (of air), and fluid friction (meteors).

* With increasing air temperatures the ignition temperatures decrease.

In certain apparatus, combustion is not continuous, and ignition must be intermittently and precisely accomplished. The most familiar example is the *internal combustion engine.* Two common variants of this engine are the spark ignition (Otto) and the compression ignition (Diesel) engines. In either case fuel and air must be mixed and raised to the ignition temperature, or above.

Combustion, being by nature a chemical reaction, is readily depicted by equations of chemical reaction which, fortunately, are quite elementary. The following equations illustrate the combustion of carbon to carbon dioxide.

Molecular reaction:	C	$+ O_2$	$= CO_2$
Combining mols:	1 mol C	$+$ 1 mol O_2	$=$ 1 mol CO_2
Combining weights:	1×12 lbs. C	$+ 1 \times 32$ lbs. O_2	$= 1 \times 44$ lbs. CO_2

Thus, each 12 lbs. of carbon require 32 lbs. of oxygen for complete combustion. Taking into consideration the fact that air has only 23.2% by weight of oxygen, a pound of oxygen is contained in 1/.232 lb. of air; consequently, in the combustion of carbon to carbon dioxide, 12 lbs. of carbon require 32/.232 lb. of air, or 1 lb. of carbon requires $\dfrac{32}{12 \times .232} = 11.5$ lbs. of air, theoretically.

Correspondingly, hydrogen needs 34.5 lbs. of air per pound, and sulfur 4.3 lbs. per lb. The air theoretically required to burn a pound of coal is therefore

$$\text{Air} = 11.5C + 34.5\left(\text{H} - \frac{\text{O}}{8}\right) + 4.35\text{S lbs. per lb. of coal.}$$

The symbols represent fractional proportions of the elements in the ultimate analysis. The O/8 correction allows for the hydrogen which is already combined.

The theoretical combustion of carbon with 11.5 lbs. of air creates the gas CO_2 as a product and retains inertly the N_2 of the air. Taking the composition of air as 76.8% N_2 by weight, the 11.5 lbs. contain 8.83 lbs. N_2. The quantity of CO_2 produced may be found by reference to the foregoing equation, which shows 44 lbs. CO_2 produced by 12 lbs. C. Obviously, a pound of C would produce $\frac{44}{12}$ or 3.67 lbs. CO_2. Assembling all quantities just calculated for this particular reaction, i.e., carbon with air,

$$1 \text{ lb. C} + 11.5 \text{ lbs. air} = 3.67 \text{ lbs. } CO_2 + 8.83 \text{ lbs. } N_2.$$

The combustion of a compound such as C_8H_{18} might be analyzed in the same way by considering separately the combustion of the carbon and hydrogen, afterwards totalling the products (this relies on the principle of conservation of mass). C_8H_{18} has molecular weight of $8 \times 12 + 18 \times 1$ or 114, hence 1 lb. of C_8H_{18} can be considered to be $\dfrac{8 \times 12}{114}$ lb. C and $\dfrac{18 \times 1}{114}$ lb. H_2.

Alternately, an equation of combining mols could be written for the compound and from it the air and products determined as follows:

Molecular reaction:	C_8H_{18}	$+ 12.5O_2$	$= 8CO_2$	$+ 9H_2O$
Combining weights:	114 lbs. C_8H_{18}	+ 400 lbs. O_2	= 352 lbs. CO_2	+ 162 lbs. H_2O
(Same per lb. fuel):	1 lb. C_8H_{18}	+ 3.51 lbs. O_2	= 3.09 lbs. CO_2	+ 1.42 lbs. H_2O
Reaction with air:	1 lb. C_8H_{18}	+ 15.15 lbs. air	= 3.09 lbs. CO_2	+ 1.42 lbs. H_2O + 11.64 lbs. N_2

TABLE 3–2. COMBUSTION PROPERTIES OF FUELS

Fuel	Molecular Weight	Molecular Equation	Pounds of Air Required per Pound of Fuel	Products of Combustion, Pounds			Heating Value, B.t.u. per Pound	
				CO_2	H_2O	N_2	Higher	Lower
C to CO_2	12.0	$C + O_2 = CO_2$	11.5	3.67	....	8.8	14,540	......
C to CO	12.0	$2C + O_2 = 2CO$	5.7	2.33 *	....	4.4	4,430	......
CO to CO_2	28.0	$2CO + O_2 = 2CO_2$	2.4	1.67	....	1.8	4,380	......
H_2	2.016	$2H_2 + O_2 = 2H_2O$	34.5	....	9.0	26.5	62,000	52,100
S	32.0	$S + O_2 = SO_2$	4.3	....	2.0 †	3.3	4,000	......
CH_4	16.03	$CH_4 + 2O_2 = CO_2 + 2H_2O$	17.2	2.75	2.25	13.2	23,850	21,375
C_2H_4	28.03	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	14.8	3.14	1.29	11.4	21,450	20,035

* Carbon monoxide. † Sulfur dioxide.

3–10. Excess Air. The theoretical quantity of air which should supply just that amount of oxygen required for chemical union with the molecules of a fuel is often insufficient for combustion under actual conditions. In the relatively short time that the fuel and air are together in a region of high temperature, the imperfection of mixing methods, and the presence of the fuel in lumps or droplets, rather than separated molecules, combine to promote incomplete combustion unless there is an excess of oxygen over that which is actually needed. This excess air is ordinarily given as a percentage of the theoretical requirement, and the amount employed is determined by the degree of technological control of combustion practiced, the surface exposure of the fuel, i.e., whether the fuel is gaseous, atomized, powdered, or lump, and the perfection of design of the furnace and burning equipment. Excess air is shown by the presence of oxygen in the products of combustion.

Coals ranging from the hard and nearly smokeless anthracite through intervening ranks down to the brown and woody lignite have their combustible elements combined in many different ways but, strangely enough, the air required to produce a thousand heat units from any coal is very nearly a

constant amount. This fact may be turned to good account in rapid determination of the air needed for combustion if the calorific value of the coal is known. Trial computations show that a pound of combustion air is theoretically required for each 1340 B.t.u. produced in complete combustion, for any coal chosen.

The excess air requirements of combustion equipment will vary. Lump coal, fired by stokers, will usually be completely burned with the use of 50% of excess air, whereas in the hand-fired stokers, up to 100% may be necessary, and undoubtedly a great deal of domestic coal firing, unaided by technological information or experience, is carried out with from 150% excess air upwards. Pulverized coal and fuel oil may be satisfactorily fired with 5 to 15% excess air. Gaseous fuels fired with the aid of well-designed burners may be so effectively mixed with air that no excess air is needed. Some internal combustion engines operate with no excess air whereas other types may use up to 100% excess.

3-11. Products of Combustion. The combustion process leads to the production of oxidized fuel elements H_2O, CO_2, and CO. As coal generally contains some sulfur, the products will include SO_2. Combustion in air as the source of oxygen inevitably dilutes the whole process with quantities of inert nitrogen. Since air is over 75% nitrogen, this gas accounts for most of the volume of the products of combustion. The necessary use of some excess air naturally leaves some oxygen also in the products. Hydrocarbon fuels (mainly liquid) often burn imperfectly in the combustion chambers of engines, leaving a residue of some hydrocarbon fuel gases in the products of combustion. The combustion of coal always results in a residue of inorganic ash, and in some instances this ash will be found intermixed with small particles of carbon or coal whose combustion has been smothered by ash.

These, then, are the products of combustion. They are the waste of a manufacturing process in which a useful commodity—heat energy—is made from two raw materials. One, fuel, is an economic goods while the other, oxygen, is usually taken (along with the unwanted nitrogen) as the free gift of nature.

Were all the heat of combustion to be utilized to heat the products of combustion, very high temperatures would result. For example, consider the reaction of carbon with pure oxygen. If it produces the fully oxidized product CO_2, the reaction is illustrated by the equations:

$$C \quad + O_2 \quad = CO_2$$

$$1 \text{ lb. } C + 2.67 \text{ lbs. } O_2 = 3.67 \text{ lbs. } CO_2.$$

As the heating value of carbon from Table 3-2 is given as 14,540 B.t.u. per lb., this quantity of heat would theoretically be absorbable as sensible

heat by the 3.67 lbs. of CO_2. The resulting temperature rise could be calculated from the equation for sensible heat:

$$Q = wc\,\Delta T,$$

in which Q = sensible heat, B.t.u.
c = specific heat, appropriate for the circumstances.
w = weight of matter which undergoes a temperature change.
ΔT = temperature change.

An approximate value for c appropriate for CO_2 at high temperatures is .27 B.t.u. per lb. per deg. F. Then

$$\Delta T = \frac{14{,}540}{3.67 \times .27} = 14{,}680^\circ \text{ F}.$$

Actually no such temperature as this would be attained since a partial dissociation of the CO_2 with attendant endothermal action would occur at some lower temperature. This dissociation takes place as follows:

$$CO_2 = CO + \tfrac{1}{2}O_2.$$

Enough of this reverse action will occur to maintain the temperature at some equilibrium value where oxidation and dissociation are equal. This varies with the pressure but in most cases is between 4000° and 5000° F. For the combustion of fuels in air (with some air in excess of theoretical requirement) where considerable heat is radiated to the surroundings, as in boiler furnaces and engine cylinders, the temperature does not rise sufficiently high to produce much dissociation; however, it is a factor limiting the combustion chamber temperature of some engines.

The combustion of coal is the common source of heat energy, the most important usage being to generate steam in boilers. Since the products of combustion of such fires are eventually conveyed to the atmosphere through the *flues* of a boiler or chimney, the products of combustion are commonly referred to as *flue gas.* All the products mentioned above are usually present in flue gas to some degree. There should be little or no carbon monoxide, for that is indicative of faulty or incomplete combustion. The oxygen content should be as low as it may be made consistent with maintenance of complete combustion. Since air is so largely composed of the inert gas nitrogen, the bulk of flue gas is also nitrogen. A typical analysis of dry flue gas by volume as produced by a coal fire, well tended, would be carbon dioxide, 12%; oxygen, 8%; nitrogen, 80%.

While poor combustion of fuel is frequently detectable by the evolution of dense clouds of smoke, smokeless combustion may not necessarily be the best

possible that may be obtained in the circumstances, since unnecessary quantities of *excess air*, or the product of incomplete combustion, carbon monoxide, may be present to render combustion inefficient, even though the flue gas is not badly colored. The flue gas content is the fireman's best indication of actual combustion conditions. For combustion test purposes, including performance runs for efficiency, checking operation of equipment, etc., complete

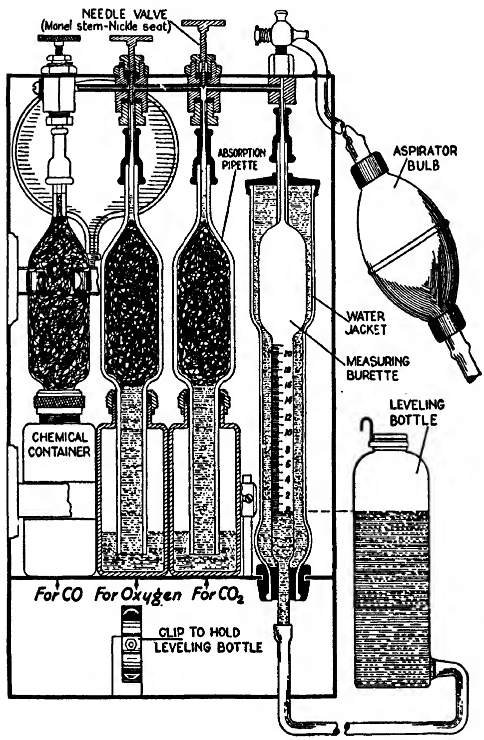


FIG. 3-3. Orsat apparatus. (Courtesy Hays Corp.)

analysis of the four principal gases in the products of combustion is required. For this analysis the Orsat apparatus is indispensable; indeed, the analysis is usually referred to as Orsat analysis. The apparatus analyzes a measured volume of a mixture of gases by the process of absorption. The volume remaining is measured, thus indicating by difference the gas absorbed. The remainder is then exposed to another reagent, which removes another gas. Upon remeasurement, the volume of that gas becomes known, and this process is continued a third time, so that the percentages of carbon dioxide, carbon monoxide, and oxygen are determined. The remainder is assumed to be nitrogen, as it, in fact, is, for all practical purposes. The analysis obviously is one by volume, and on account of the construction and use of the apparatus, it is made at atmospheric pressure and temperature.

PROBLEMS

1. A sample of coal is tested for its proximate analysis, which is found to be fixed carbon, 59%; volatile, 24%; moisture, 8%; ash, 9%. Compute the percentage analysis of the combustible part of this coal by increasing the percentage of the combustible items proportionately until they will total 100%.

2. What is the proximate analysis of the combustible part of Pennsylvania Anthracite (Table 3–1)? Note method suggested in Problem 1.

3. The coal of Problem 1 is stated to have a heating value of 12,540 B.t.u. per lb. What heating value is produced by a pound of the combustible? Assume that the combustible can be freed from the incombustible items. Of course all the heating value of the coal resides in its combustible fraction.

4. A sample of oven-dried coal yields the following ultimate analysis: C 78.1, H_2 4.3, O_2 3.1, S 2.6, Ash 11.9. Upon exposure of a .550-gm. sample to the weather for several days the weight is found to be .571 gm. Construct an analysis of the moist coal: (a) with moisture a separate item; (b) with moisture broken up and added to H_2 and O_2.

5. Rewrite the ultimate analysis of Pocahontas coal (Table 3–1) so as to show moisture a separate item. Then rewrite it again, accounting for all the remaining oxygen by "water of crystallization." Why doesn't the moisture of the proximate analysis include this water of crystallization?

6. Repeat Problem 5 for Indiana Bituminous coal.

7. How much carbon is there in the hydrocarbon compounds of a pound of Pocahontas coal (Table 3–1)? Assume that all of the volatile that is not "free" hydrogen is carbon.

8. The Indiana Bituminous coal of Table 3–1 has 38.59% volatile. This is a compound of carbon, water of crystallization, nitrogen, sulfur, and "free" hydrogen. All oxygen in the fuel is accounted for either in free moisture (of the proximate analysis) or water of crystallization. Find the carbon in the volatile by computing first the other four constituents of the volatile. Find an alternative solution for carbon in the volatile and demonstrate that the two methods agree.

9. Repeat Problem 8 for Pennsylvania Anthracite.

10. Find by calculation the heating value of CH_4. Explain why this differs from the value shown in Table 3–2.

11. Compute the heating value (higher and lower) of a coal whose analysis is C 70, H_2 5, O_2 8, N_2 1, S 3, Ash 13.

12. Compute the higher and lower heating values of Indiana Bituminous coal (Table 3–1). A calorimeter test on this coal gives 11,212 B.t.u. Discuss reasons for any difference.

13. The higher heating value of acetylene (C_2H_2) by calorimeter is 1455 B.t.u. per cu. ft. at 14.7 psi. and 60° F. Resolve a pound into the component parts of carbon and hydrogen, and calculate the heating value by formula. At the pressure and temperature stated a mol of any gas occupies 380 cu. ft. What is found to be the computed heating value per cubic foot? What kind of reaction is the breakdown of C_2H_2 into C and H_2? Does this mean that the acetylene molecule is relatively stable or unstable?

14. In the products of combustion of a pound of coke (C 87.0, Ash 13.0) there are .05 lb. carbon, 1.01 lb. carbon monoxide, and an undisclosed quantity of carbon dioxide. How much heat was released?

15. How much carbon dioxide appeared in the products of combustion for the conditions given in Problem 14?

16. Compare the energy released when one pound of hydrogen is burned (oxidized to H_2O) with that released if one billionth of atoms in the pound could be disintegrated into energy. (As will be shown in Chapter 4, a B.t.u. is equivalent to 778 ft. lbs. of work.)

17. How does the calculated heating value of a pound of Pennsylvania Anthracite compare with that of a pound of gasoline?

18. What is the weight per gallon of oil of 28° Baumé? Water weighs 8.33 lbs. per gallon.

19. A barrel of oil contains 42 gallons. What is the weight of a barrel of 30° Baumé oil?

20. A measuring vessel when filled with water to a certain point weighed 2.67 lbs. When filled with oil to the same mark it weighs 2.19 lbs. Empty weight is exactly three-quarters of a pound. Estimate what the Baumé reading of the oil should be.

21. A certain 25° Baumé oil analyzes 84.4% carbon and 15.6% hydrogen. Calculate the higher heating value: (a) in B.t.u. per lb.; (b) in B.t.u. per gallon.

22. What is the lower heating value of pure hydrogen, B.t.u. per cu. ft. at 14.7 psi. and 60°? (Molal volume = 380 cu. ft. at these conditions.)

23. Estimate the higher heating value of a cu. ft. (standard condition) of typical producer gas. See Table 3–2 for heating values for the individual gases.

24. From text description construct a diagram of a gas calorimeter. Label fully.

25. Gas flows through a Junkers calorimeter for a half hour, during which 0.675 cu. ft. was registered on the precision gas meter. During the same time water was flowing steadily through the calorimeter at the rate of 3.56 lbs. per minute, and absorbing heat so that its temperature rose from 62.2° to 68.1°. Neglecting all corrections (which are necessary to validate an actual test) estimate the heating value of the gas, B.t.u. per cu. ft., on the basis of the law of conservation of energy. Is this higher or lower heating value? Why?

26. A good test rate of gas consumption in a certain gas calorimeter is .02 cu. ft. per min. If the water temperature rise is to be approximately 8° and the gas will test about 900 B.t.u. per cu. ft., how much water (gallons) should be accumulated in a storage tank in order to provide for four 15-min. tests?

27. A bomb calorimeter is used to test a sample of coal for heating value. A sample weighing 1.26 gm. is placed in the pan and 2.5 in. of fuse wire used for ignition. The water surrounding the bomb weighs 12.5 lbs. Heating value of wire is 1.3 B.t.u. per in. Temperature rise of water was 2.41° F. Neglecting all corrections except fuse wire, compute heating value of this sample. Is this higher or lower heating value? Why?

28. Suppose it is planned to test kerosene (heating value approximately 18,000 B.t.u. per lb.) in the calorimeter whose size is partially described in Problem 27. How much kerosene (grams) could be burned in one test without the temperature rise of the water bath exceeding 5°? Neglect all corrections except fuse wire.

29. Draw a sectional diagram of a common match. Label same fully.

30. Diagram an arrangement by which the sun will furnish ignition of cellulose.

31. The proper amount of oxygen for ideal complete combustion is mixed with finely powdered carbon at 60° F. The specific heat of the mixture is given as .25 B.t.u. per lb. per deg. F. How much heat will be added before the carbon and oxygen will spontaneously combine to form CO_2?

32. A sample of coal weighing 4.55 gm. is heated at 220° for an hour to drive off free moisture. Upon re-weighing it is found to weigh 4.12 gm. Another sample of the same coal is weighed out and heated to a temperature sufficient to drive off all the volatile gas. It weighed 4.42 gm. before and 3.27 gm. after heating. Finally when a 10.22-gm. sample of this coal was burned to ash carefully so no ash was lost. the residue weighed 1.17 gm. Compute the proximate analysis from the data given.

33. Write combustion reactions for CH_4 with O_2. Derive equations of mols based on 1 mol of CH_4. Explain why this is equally correct as an equation of volumes, based on 1 cu. ft. CH_4 provided all gases are measured at the same pressure and temperature.

34. Write the combustion reaction of sulfur with the theoretical quantity of oxygen and derive an equation of weights per pound of sulfur.

35. Prove that the combustion of a pound of carbon monoxide produces 1.57 lbs. of carbon dioxide.

36. Write the reaction between H_2 and O_2. How many pounds of air containing 23.2% O_2 by weight are required, theoretically, to burn a pound of hydrogen?

37. Calculate the pounds of air required for the ideal combustion of one of the fuels listed in Table 3–1.

38. The combustion of a certain liquid fuel rocket is supported by alcohol (CH_4O) and liquid oxygen. Experience indicates that no excess oxygen is needed for this type of combustion. Find the total weight of propellent to be carried by a rocket which is designed to carry 7500 lbs. alcohol. Compare your calculations with data for the German V2 rocket.

39. A rocket which is to burn C_8H_{18} with the ideal quantity of liquid oxygen is to be designed to expel 100 lbs. per sec. of products of combustion for a period of 1.25 minutes. Specify the pounds of gasoline and liquid oxygen required at the start of the shot.

40. It is found that there are 2 lbs. O_2 in the products of combustion of 1 lb. oil whose analysis is C 85, H_2 15. There is no incomplete combustion. What per cent excess air does this represent?

41. How many pounds of N_2 should the products of combustion of the oil mentioned in Problem 40 contain if it were burned with 100% excess air? How much O_2?

42. A pound of carbon is burned incompletely, leaving 0.2 lb. CO in the products. What potential heating value is thus lost?

43. How many cubic feet of air at 14.7 psi. 60° F are required for the ideal combustion of a cubic foot of butane measured at the same condition? Air contains 20.9% of oxygen by volume.

44. The crude oil mentioned in Table 3–1 is to be burned with sufficient air for complete combustion. It is estimated that 85% excess air will be adequate. How many pounds of air will be used per pound of oil? How many cubic feet at 60°?

45. What is the theoretical maximum temperature of combustion of Indiana Bituminous coal burned with 80% excess air? Assume $c = .25$.

46. Calculate the ideal air-fuel ratio of gasoline.

47. How much excess (percentage) air must be used in the combustion of fuel oil (C 84.5, H_2 15.5) to hold the maximum theoretical temperature of combustion to 2000°. Air temperature 65° F. Assume $c = .26$.

48. A flue gas analysis was made beginning with 100 parts by volume (see Figure 3–3) of the gas. After absorption of CO_2 the volume was 89.3 parts. After absorption of O_2 the volume had shrunk to 81.7 parts, and finally after absorption of CO the volume was 81.1. Make out the Orsat analysis for this sample of gas.

49. An Orsat apparatus like that diagrammed in Figure 3–3 is used on a sample of boiler flue gas. The burette was originally full of the sample to be measured. After being passed in turn through each of the absorption pipettes from right to left (being brought back to the burette for measurement between each absorption) the following readings of water level in the stem of the burette were recorded: 9.5, 19.7, 19.8. Make out the Orsat analysis from these data.

50. If carbon is burned ideally in air it produces CO_2 and N_2 in the products of combustion.

a. How many pounds of CO_2 and N_2 would the theoretical combustion of 1 lb. C yield?
b. How many cubic feet of each of these gases would there be at standard atmospheric conditions (employing the principle of the molal volume to convert weight into volume)?
c. Write an Orsat analysis of the products. This represents the maximum CO_2 possible in products of combustion.

CHAPTER 4

Energy in Action

4-1. Principles of Conservation. If a certain system contains a mass of m grams of matter and an energy of e ergs, its true "inertial mass" is $m + (e/c^2)$, wherein c is the speed of light.* Dynamic energy possesses an inertial mass itself of e/c^2. However, the numerical magnitude of c^2 is so enormous that, unless release of subatomic energy (which is also enormous) is involved, the inertial mass of energy is negligible.†

There are two practical derivatives of this, i.e., (1) the principle of "*conservation of mass*" and (2) the principle of "*conservation of energy*." $m_2 = m_1$ expresses the first principle, where these represent the mass of an isolated system at two successive intervals of time. It was long thought to be absolutely true, but we now know that in some cases (as in atomic fission) m_2 can be detectably different from m_1. The second principle follows from the first. Conservation of energy may be expressed thus: $e_2 = e_1$. It is valid for all applied energy derivations except those including atomic energy. In this equation, energy e is the sum of all the energy present in an isolated system. It can be expressed in units of any of the forms of energy as long as the same unit is employed throughout the equation. Since all forms of energy are mutually interchangeable in direct ratio, no great difficulty is experienced in setting up a conservation-of-energy equation even though energy may consist of two or more different kinds existing simultaneously in the system. But, we shall have to know something about the ratio of interchange.

That energy is interchangeable between its different forms one should not for the moment doubt, for all around us in the modern world the transformations can be seen in action. Heat goes over into work in engines; and work into heat wherever friction exists. Electrical energy produces heat in stoves and various heating appliances. Heat may produce radiation energy by raising a substance to incandescence. Radio transmission depends on electrically produced radiation. Work is transformed into electrical energy in generators; vice versa in electric motors. So it goes; we live in the midst of energy transformations of all kinds. They have become an intimate necessary part of civilized existence. It is now pertinent to inquire at what ratios of energy these transformations occur.

* See page 18.

† Where it is not negligible, $m_2 + (e_2/c^2) = m_1 + (e_1/c^2)$ in an isolated system.

4–2. Energy Equivalence. Late in the eighteenth century Count Rumford upset the old theory that heat was an invisible fluid and showed it to be a form of energy. Then (1840) James Joule worked out, experimentally, the numerical relation between work and heat. Joule converted the mechanical energy of a descending weight into heat energy by causing it to churn water in a calorimeter and so heat that water by the fluid friction evolved. This experiment resulted in establishing a *mechanical equivalent of heat.* Joule found that 774 ft. lbs. of work were consumed to create a B.t.u. Subsequent experiments of greater refinement have raised this slightly and the approved value now stands at 778.3 ft. lbs. per B.t.u. The common symbol for this factor is J.

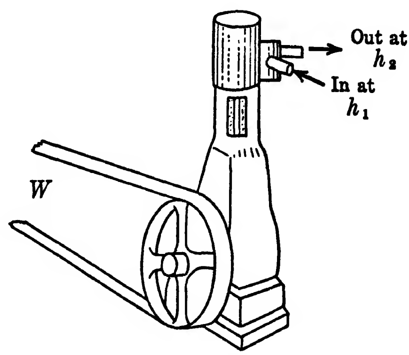


FIG. 4–1. Steam engine.

To see how this factor is used in the $e_2 = e_1$ equation, consider a steam engine through which a pound of steam passes. It enters the engine with an enthalpy h_1 and a small negligible amount of kinetic energy. It emerges, after undergoing an expansion, with a smaller enthalpy h_2. Some work has been created from the heat and is now available at the pulley. If the engine is assumed to be well insulated against heat leakage to the surrounding atmosphere, and has negligible internal friction, then

$$h_1 = h_2 + \frac{W}{J}.$$

The foot-pounds of work, W, must be divided by J so all terms of the equation will have the same dimensional unit. In this case the unit is B.t.u.'s, but it could have been foot-pounds as the following equation evidences:

$$Jh_1 = Jh_2 + W.$$

Imagine that the steam is flowing through this engine at the rate of 9.5 lbs. per min. and that calorimetric measurements on samples of it before and after expansion give it enthalpies h_1 of 1175 B.t.u. per lb. and h_2 of 985 B.t.u. per lb. Then the work produced is $J(1175 - 985)$ ft. lbs. per lb. steam, or $9.5J$ $(1175 - 985)$ ft. lbs. per minute. By recalling definitions of horsepower from Chapter 1, we conclude that this continuous flow of steam permits the engine to develop 42.6 hp.

$$\text{Power} = \frac{9.5 \times 778.3(1175 - 985)}{33{,}000} = 42.6 \text{ hp.}$$

Next, the heat developed at the brakes in stopping a 2200-pound automobile travelling at 30 mi. per hr. is calculated for another application of the mechanical equivalent of heat. Let e_1 be the kinetic energy of the automobile just before the brakes are applied. At the instant the automobile comes to rest the kinetic energy has disappeared but heat has developed at the brake drums. Call this heat Q. Assuming a level road, also negligible air and rolling resistance, $e_1 = JQ$.

$$Q = \frac{e_1}{J} = \frac{\frac{1}{2}mv_1^2}{J} = \frac{\frac{1}{2} \times \frac{2200}{32.2}\left(\frac{30 \times 5280}{3600}\right)^2}{778.3} = 85.6 \text{ B.t.u.}$$

The conversion factor between work and electrical energy is sought next. A flow of electrons of a coulomb per second is an ampere. When this is impelled by a potential of one volt the electrical power is one volt ampere, which is also called a *watt*. This is also a Joule per second. Electric energy can therefore be measured in Joules which is the same as watt seconds. Since, as was mentioned in Chapter 1, a foot-pound is equivalent to 1.355 Joules, a watt second becomes 1/1.355 or 0.7375 ft. lbs. This establishes a relationship between the common units of electrical energy * and mechanical work. Now the ratio between the ordinary commercial unit of electrical energy, the kilowatt hour, and heat can be calculated. A kilowatt hour is $60 \times 60 \times 1000$ watt seconds. The heat equivalent is $60 \times 60 \times 1000 \times .7375/778.3$ or 3412 B.t.u. A kilowatt hour is the same amount of energy as 3412 B.t.u.'s. The power of one kilowatt represents the production or consumption of energy at the rate of 3412 B.t.u. per hr. The reader may parallel this by demonstrating that a horsepower hour is equivalent to 2545 B.t.u.'s.

4–3. Energy Transfer. Before proceeding to further consideration of energy transformation, attention is directed to an action where the form of energy is not altered, but it is transferred from one locality to another. The thermal efficiency with which transfers can be effected is usually quite high—much higher than can be expected in some energy transformations. As heat is the least organized form of energy, the losses, if any, will usually appear in that form.

Mechanical energy is transferred by mechanisms such as belts, gears, or shafts while electrical energy is carried by conductors (transmission lines) and by electromagnetic radiation. Mechanical transfer losses usually appear as frictional heat, electrical losses as resistance heat. The transfer of heat energy from one point to another is an event of common occurrence in industry and in systems built to generate mechanical power from fuels as a source

* Another unit of importance to certain technical fields is the *electron-volt*. Discussions of atomic energy are often framed in a large multiple of this unit, the *Mev* or million electron-volts.

of energy. Full justice cannot be paid to the technology of heat transfer in this book, but a brief introduction is contained in the next section.

4–4. Heat Transfer. The transfer of heat from one place to another is one of the common needs of industry. Much study has been given to the action of heat transfer and quantities of industrial equipment have been built and used to secure it in one way or another. Heat can be transferred by three methods: by conduction, where the heat must diffuse through solid material or stagnant fluid; by convection, where the heat is carried from one point to

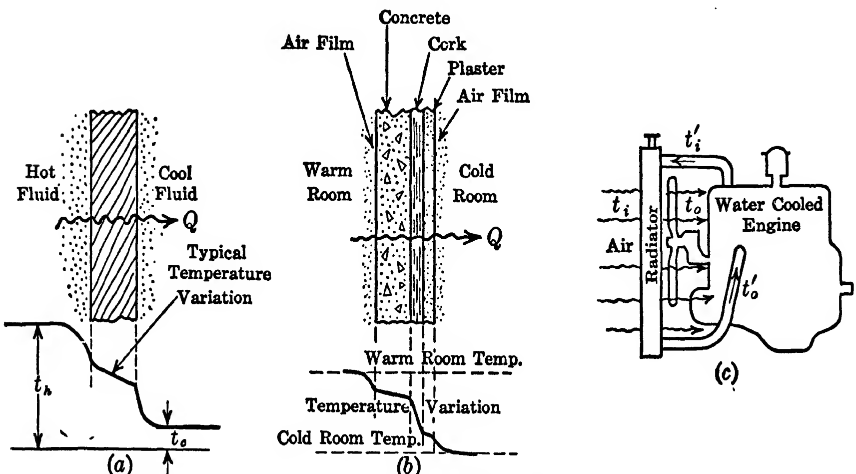


FIG. 4–2. Illustrations of heat transfer.

a. Section of partition bathed with hot and cool fluids.
b. Typical multiple layer heat insulating wall of cold storage room.
c. Case of heat transfer by convection with both fluids varying in temperature.

another by actual movement of the hot substance; and by radiation, where heat is transferred as radiant wave energy. Some of the more common cases of heat transfer are:

1. Radiation from flame and other luminous material to absorptive surfaces such as boilers, cylinder walls, etc.
2. Radiation from heat generators such as drying lamps.
3. Convection of heat by products of combustion.
4. Convection of heat from hot surfaces by a fluid under either free or forced circulation.
5. Conduction of heat through the walls of heat exchangers, condensers, boilers, etc.
6. Conduction of heat through building walls, pipe coverings, and other so-called "heat insulators."

Conduction. The passage of heat energy through a substance by the process of conduction is a molecular phenomenon. Fast-moving molecules in the

hotter region yield up energy by collision with their slower moving neighbors who thereby are energized sufficiently to continue a like transfer to their less active companions. In stagnant fluids this unbalanced kinetic activity may, in addition, lead to some diffusion of fast-moving molecules into cooler regions. When a solid partition is conducting heat from one fluid to another, as in the figure on page 77, a thin layer of fluid remains stagnant against each face of the partition and becomes, in effect, another conducting layer outside of which the free stream fluid temperature exists. Although this film is extremely thin, its relative resistance to heat flow may be high compared to the solid partition. This is especially true in heat exchangers where the solid wall material is metal of good conducting properties. Although film conductance is difficult to study or evaluate, being under the control of several variables (i.e., fluid velocity, viscosity, density, turbulence), its importance to conductive heat transfer is considerable.

The fundamental equation for heat transferred by conduction (Fourier's Law) is:

$$Q = \frac{AK\Delta T}{d},$$

in which Q = Heat energy conducted per unit time.
A = Conducting area, measured normal to direction of heat flow.
ΔT = Temperature difference forcing the heat flow.
d = Thickness of the conducting layer.
K = *Coefficient of conductivity* of the material. The dimensional units taken for K determine the necessary physical dimensions of the other factors. In this book Q will be B.t.u. per hr., A sq. ft., ΔT deg. F, and d ft.

This law of conduction may also be stated:

$$Q = UA\Delta T,$$

in which U is the *conductance*. It is obvious from the above that K and U are related as follows:

$$U = \frac{K}{d}.$$

The reciprocal of conductance is thermal resistance. It is the *resistance* that is additive when several conducting layers lie between the hot and cool regions. In a multiple layer partition having an overall conductance, U and layer conductances U_1, U_2, etc.,

$$\frac{1}{U} = \frac{1}{U_1} + \frac{1}{U_2} + \frac{1}{U_3} \cdots, \text{ etc.}$$

TABLE 4-1. TYPICAL CONDUCTIVITIES OF SOME COMMON MATERIALS

Material	Temperature, deg. F	Conductivity, B.t.u. per hr. per sq. ft. per deg. F per ft. thickness
Air	50	.014
Aluminum	100	122.0
Asbestos, Loose	400	.05
Brick, Fire	2000	.9
Brick, Red	100	.27
Concrete, Stone	60	.6
Copper	100	220.0
Corkboard	80	.025
Insulating Wallboard	60	.03
Magnesia, Powdered	400	.06
Rock Wool	90	.023
Rubber, Soft	60	.10
Steel	100	25.0
Water	100	.35
Wood, Pine	60	.07

Convection. Transfer of heat by *convection* implies a carrying medium travelling from a hot to cool region. While this might be a solid (conveying, each trip, heat in the amount of weight $\times$ specific heat $\times$ ΔT), the important cases of convection arise in connection with fluids in steady motion as the carrying media. Dependent on whether the fluid is under forced, controlled motion or is moved by natural density differences caused by the heating itself, the convective process is termed "forced" or "free." Heat transfer in forced convection is more readily analyzed and predicted than in free convection, although the latter is important in the liberation of heat from steam "radiators," hot walls, pipe lines, and other static hot surfaces about which circulate "free" fluid currents.

Sometimes a case of heat transfer by convection can be solved analogous to conduction if a *heat transfer coefficient* (similar to conductance U) can be determined. If U' is such a coefficient, then $Q = U'A\Delta T$ is the heat transferred by convection between a surface of area A and a fluid when the difference in temperature is ΔT. The value of U' may depend on several variables such as fluid velocity, viscosity, shape of the surface A, etc. Many cases have been studied so that information is available for almost any practical problem. It is observed that the coefficient U' can be handled like a conductance in determining the overall coefficient where convection is in series with conduction.

Thermal Radiation. Radiant energy is that which can be transferred through space in the absence of any physical carrying medium. All bodies that are not at absolute zero emit radiation as the result of their kinetic atomic state. It derives from the electrons as wave energy, with a wavelength and frequency dependent on the substance and its temperature. The range of wavelengths for visible light and heat is between 4 and 7 ten-thousandths of a millimeter. Shorter wavelength energy is ultra-violet light; longer is infra-red heat. An ideal "black body" is one which absorbs all wavelengths of radiant energy equally well. Ordinary surfaces possess this property imperfectly, but can be made to simulate a black body surface if a dull black coating is applied.

Radiant energy travels in straight lines from the source of emission; therefore, to transmit it out of sight of the radiator requires a reflector to deflect it or a suitably placed re-radiator (such as an incandescent wall) to re-direct it. In ordinary surroundings, bodies not only emit, but also receive, radiation. The net energy transferred depends on the relative rate of emission and absorption, and this, in turn, on the relative temperatures of the body and its surroundings. Many have studied this relationship, among them Stefan and Boltzmann, who found that the energy transferred varies as the fourth powers of the two temperatures. In symbols,

$$Q = CA(T_1^4 - T_2^4),$$

in which Q = Heat energy transferred per unit time.
A = Surface area of the radiating body.
T_1 = Absolute temperature of the radiating body.
T_2 = Absolute temperature of the receiving body.
C = A radiation coefficient, dependent on the nature of surface A.

It is possible to set up controlled laboratory radiation between simple plane surfaces and determine therefrom accurate coefficients to incorporate into radiation equations. However, the radiation of heat from furnace gases, consisting of non-luminous gases, luminous carbon particles in flame, ash globules, etc., to the walls and tubes of a steam generator in commercial operation at variable load, is another matter. Here, empirical data which are gathered and interpreted from field tests on similar equipment are usually consulted.

Mean Temperature Difference. In the case of heat transfer from a fluid medium on one side of a conducting partition to another one on the other side, it is likely that one or both fluid temperatures will be variable. If a fluid is undergoing evaporation or condensation it can change its heat content with no change of temperature, but otherwise heat transfers mean temperature changes. Since the temperature difference is of primary importance to heat transfer through a conducting partition, its mean or average value should receive careful study. Consider a case of heat transfer through a partition

illustrated by a tube surrounded by a medium at t', and through which flows another medium which rises from t_i at inlet to t_o at outlet of the tube. The arithmetical mean temperature difference is the simple average of the terminal temperature differences.

$$\text{Arithmetical } m.t.d. = t' - \left(\frac{t_i + t_o}{2}\right).$$

The arithmetical *m.t.d.* is only an approximation of the true thermal *m.t.d.* Thermodynamic theory yields the following equation for true *m.t.d.*

$$\text{Thermal } m.t.d. = \frac{t_o - t_i}{\log_e \dfrac{t' - t_i}{t' - t_o}}.$$

If the first medium changed in temperature from t'_i to t'_o instead of remaining constant, then the

$$\text{Thermal } m.t.d. = \frac{(t'_i - t_o) - (t'_o - t_i)}{\log_e \dfrac{t'_i - t_o}{t'_o - t_i}}.$$

Aiding the transfer of heat are such conditions as material of good conductivity, high driving temperatures, good radiation coefficients exemplified by high surface emissivity, rapid circulation of convection currents, etc. Opposing heat transfer are the usage of covering materials of purposely low conductivity, the polishing of surfaces to reduce radiation emissivity, the formation of stagnant fluid layers or the restriction of circulating currents, and similar action. Just as some conditions call for design and operation which will promote rapid heat transfer, so may others require the purposeful obstruction of heat flow. Cases of the latter type are illustrated by the insulation of buildings, of heat storage tanks, of refrigerating rooms, and of pipe lines.

Bare surfaces at temperatures considerably above atmospheric lose much heat to the atmosphere. The B.t.u. per hour loss from bare steam pipe may not, on first thought, seem to amount to much but, if it be remembered that this loss is nearly steady the 8760 hours of the year (unless the pipe is in intermittent service), and that often the B.t.u. so lost are high potential heat and therefore more valuable than the average B.t.u. in a pound of steam, it will be understood why practically every hot pipe in the modern plant or factory is covered. Cold pipes are also insulated to keep heat out. Insulation for this service is common in refrigeration plants.

Insulating materials are usually characterized by light weight. Nonconducting properties of commercial insulations seem chiefly to be derived from the presence of dead air cells they contain. The materials most commonly used are asbestos, "magnesia" (magnesium carbonate), cork, hair felt,

wool felt, rock wool, and diatomaceous earths. Most commercial insulations are either built up from corrugated asbestos paper, or laminated asbestos paper artificially roughened to produce air spaces, or are molded, or felted with asbestos, etc. A very common and effective insulation for temperatures up to 600° F is the molded "85% magnesia," so called because it is 85%

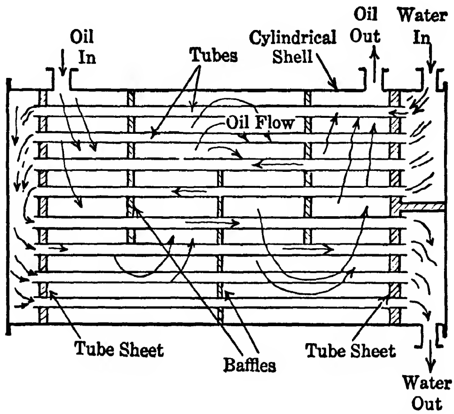


Fig. 4–3. Typical shell and tube heat exchanger (lubricating oil cooler).

carbonate of magnesium and 15% binder. Painting with aluminum or bronze paint will greatly decrease the radiation losses, as will also silvering or polishing the surface.

Example 1: A square steel bar 2 in. × 2 in. × 24 in. is heat insulated all over with the exception of its ends, one of which is in a hot liquid, the other in a cold liquid. The temperatures of the two ends are 150° F and 40° F, respectively. How fast is heat being transferred from the hot to the cold liquid?

The coefficient of conductivity from Table 4–1 is suitable for the temperature range here involved.

$$K = 25, \quad \Delta T = 110°, \quad A = \frac{2 \times 2}{144}\text{ sq. ft.} \quad d = \tfrac{24}{12} = 2\text{ ft.}$$

$$Q = 25 \times \tfrac{1}{36} \times \tfrac{110}{2} = 38.2\text{ B.t.u. per hr.}$$

Example 2: Assume that the wall shown in Figure 4–2b is made of 3 in. concrete, 1 in. cork, and 1 in. plaster. Cold room temperature is 20° F, warm room 80° F. Air film *conductance* is assumed 1.8 B.t.u. per sq. ft. per hr. per deg. What is the heat leakage per sq. ft. of wall area? Using conductivity data from Table 4–1,

$$\text{Conductance of concrete} = \frac{.6}{\frac{3}{12}} = 2.4.$$

$$\text{Conductance of cork} = \frac{.025}{\frac{1}{12}} = .3.$$

$$\text{Conductance of plaster} = \frac{.6}{\frac{1}{12}} = 7.2 \text{ (assumed same conductivity as concrete).}$$

Obtain the overall resistance by adding resistances from 80° to 20° regions.

$$\frac{1}{U} = \frac{1}{1.8} + \frac{1}{2.40} + \frac{1}{.3} + \frac{1}{7.2} + \frac{1}{1.8} = 5.0.$$

$$U = .20 \text{ B.t.u. per hr. per sq. ft. per deg.}$$

$$Q = .20 \times 1 \times (80 - 20) = 12 \text{ B.t.u. per sq. ft. per hr. leakage.}$$

Example 3: Lubricating oil of specific gravity .91 is pumped through a shell and tube heat exchanger and cooled from 145° to 90° F. The heat is transferred to water which is allowed to rise in temperature from 65° to 85° F. How many gallons per minute of water should be circulated through the unit when cooling 420 gallons lubricating oil per hour? Specific heat of oil is to be taken as .55 B.t.u. per lb. per deg.

The lubricating oil weighs .91 as much as water (which weighs 8.33 lbs. per gal.). Heat to be transferred out of the oil *per minute* is:

$$Q = \tfrac{420}{60} \times 8.33 \times .91 \times .55(145 - 90) = 1608 \text{ B.t.u. per min.}$$

Assume specific heat of the water to be exactly 1. Then, the required water flow is:

$$\text{Water quantity} = \frac{1608}{1 \times 8.33(85 - 65)} = 9.65 \text{ gal. per min.}$$

4–5. Heat-work Transformation. This transformation is easily accomplished if the direction of action is work → heat. All the resources of engineering and science cannot completely prevent its occurrence where kinetic energy exists, since friction cannot be altogether eliminated from mechanism and friction generates heat. On the other hand the heat → work transformation has to be forced. *Heat power systems* of considerable complexity and cost must be used to obtain it. The systems in use nowadays are, in part: *

Steam power plants.
Internal combustion engines.
Gas turbines.
Rockets, and other forms of thermal jet propulsion.

In these systems two basic schemes are employed to obtain the transformation wanted.

Scheme No. 1. Molecular bombardment of a moving wall creates a force which, if allowed to move the wall against a resistance, produces mechanical work. This is the *piston and cylinder* system, synonomous with "positive displacement" and "reciprocating engine." An expansible medium, gas or vapor, is used to furnish the molecular bombardment. Since high pressures and high temperatures of the gas or vapor contribute to density and intensity

* Reversed heat → work systems, called heat pumps, are exemplified by refrigeration systems. Work is converted into heat but at the same time lifts additional heat with it from a lower to a higher temperature.

of molecular bombardment, they are employed so that the system may be adequately energized and yet have as small a physical dimension as possible. Both gas and vapor pressures may be produced by the transfer of heat to the fluid and thus a system for heat → work transformation is created. Illustrating this principle, Figure 4-4 shows a piston and cylinder. The piston forms the movable end, or wall, of the cylinder. Imagine the enclosed space to be filled with a gas at a high pressure and temperature. Each molecule is darting hither and thither with great energy. Periodically (probably many thousand times each second) it strikes the piston face. The push given the piston and transmitted along the piston rod is the aggregate of these elastic impacts. If the piston then moves, pushing back the external resisting force F, work is performed. Where did this work come from? It would be safe to opine that it came from energy contained in the gas. That answer is correct but unimaginative. Endeavoring to visualize the molecular action, we arrive at some such explanation as this. Should the molecule strike and elastically rebound from a motionless piston it would retain all its kinetic energy of motion.. But when the piston is also in motion in the same direction, the brief instant of contact allows transfer of energy in the amount of the contact pressure times the distance travelled during contact. Like two freight cars freely rolling on a track in the same direction, the faster one rapidly overtaking the slower, the collision transfers energy. The kinetic energy of the gas molecules is random but when some of it is transferred to the piston it is organized into a unidirectional force. Thus not only was there a transfer, there was a transformation, an increase of *organization* of energy.

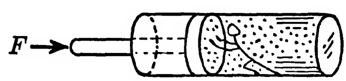


FIG. 4-4. Piston and cylinder.

The translatory speed of the molecule is enormous compared to piston speed and its mass is minute. Consequently it rebounds from the piston with almost—but not quite—the same energy it had to begin with. However, repeated contacts of it and all its companion molecules on the moving piston take a toll of the molecular energy the gas originally contained. The external evidence of this is a decrease of temperature of the gas (unless some external source is continuously replenishing the gas with more energy). This somewhat elaborate pictorization of the transformation of heat into work is given in order that the action may be compared with the second scheme of accomplishing this transformation.

Scheme No. 2. By allowing molecules to yield more in one direction than in another under the action of intermolecular collisions the whole mass of the medium may be set in motion in a desired direction. This is the principle of the *jet*. It accomplishes the heat → work conversion of "turbine" machines and has various other useful applications, including all forms of jet propulsion. The means of consummating this change is comparatively simple—a nozzle

Almost any orifice could serve as a crude nozzle, but for best results the size and shape must be carefully determined by the thermodynamic principles of nozzle flow. The following is a brief exposition of the supposed action of molecules in a jet. To form a jet the pressure at the exit from the nozzle must be less than the entrance pressure. On account of their thermal state, the molecules of gas around the entrance to the nozzle have random velocities in all directions. Collisions between molecules are of a lower order at lower pressures; consequently, a molecule moving through the nozzle in the direction of decreasing pressure will experience fewer unfavorable collisions than one trying to travel oppositely. In this manner the body of molecules, which is the gas,

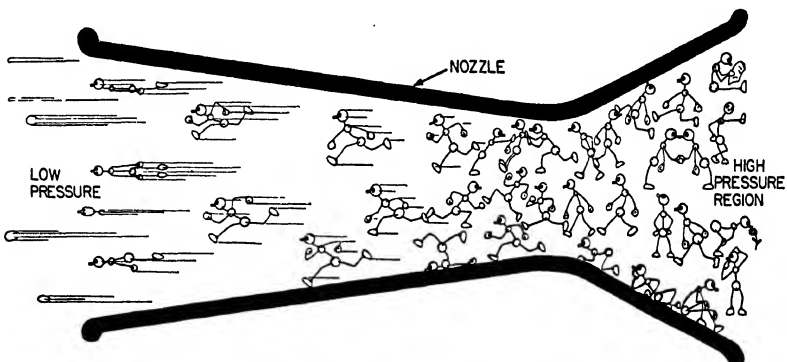


FIG. 4-5. Jet formation. The molecule-men get organized.

is set into motion through the nozzle. As the molecules will rebound from others in motion in the same direction, with lowered energy, the internal energy of the gas in the jet is diminished. The same would be true of a vapor. Part of the heat energy is organized and converted into kinetic energy of motion of the gas or vapor in the jet. Whereas the mechanical work appeared as a moving force in *Scheme 1*, here it is the external kinetic energy of a fluid in motion. The rapidly moving fluid streaming from the nozzle is the "jet." The nozzle should guide the jet in the desired direction and be of the proper flare to permit natural adiabatic expansion. To obtain work at a rotating shaft something must be provided to change the kinetic energy of the jet into a rotating torque. However, that is a mechanical problem of energy *transfer*. The nozzle achieves the *transformation* from heat to work.

Gas and steam turbines are heat → work machines (all such devices may be called *prime movers*) containing nozzles for the transformation, and bladed wheels for the transfer of energy. Supplied with an expansible fluid under sufficient pressure and temperature, the turbine will convert a portion of the heat energy into work, available at a rotating shaft, and reject the remainder as "unavailable heat." There will be more of turbines, later.

Unfortunately, only a small part of the heat energy possessed by a working medium can be converted into work energy. This is a natural result of the action of the "Second Law." * The higher the temperature at which heat energy exists above some datum temperature, such as that of the atmosphere, the ocean, etc., the more of that energy is available for conversion into work. Were the heat engine able to function perfectly under the working conditions prevailing, its thermal efficiency would be identical with the "*availability*" of the heat for transformation into work. This statement introduces the idea that one B.t.u. is not as good as another B.t.u. for working purposes. At first thought this may seem unnatural, but the circumstance is much like that of a gallon of water which may produce work in a water turbine. The gallon of water may be pent up behind the dam or it may be running merrily down the tailrace. But there is an obvious difference in the two states, for as all of us know, "the mill will never turn with the water that is past." The thing which gives the water "availability" for doing work is the pressure or "head" it possesses over the final or tailrace level. Similarly *temperature gives heat energy its availability.*

4–6. Thermodynamic Cycle Diagrams. Prime movers receive heat at high temperature, convert as much of the available part into work as possible, and discharge the remainder at low temperature. Heat pumps receive work from an external source, remove heat from a low temperature region, and discharge it to a higher temperature. The nature of the heat pump is that of a reversed prime mover. Theoretical power cycles could, if reversed, serve as heat pumps; however, actual gas compressor and refrigeration cycles are not exactly reversed power cycles because it has been found that certain modifications yielded more suitable operation.

The cycle of a displacement type of prime mover is conveniently drawn on the P-V plane because piston position can definitely fix cylinder volume. From the cycle drawn in Figure 4–6 as an example, we can obtain the following information about the cycle. Beginning at point a, which represents one extreme of piston position, the piston travel produces an expansion shown by line ab. The volume b corresponds to the other extreme piston position, and $V_b - V_a$ is the piston displacement in cubic feet. The exact manner of pressure variation between a and b depends on the work-heat transfer to or from the fluid during the process. Additional processes bc, cd, and da complete this particular *cycle*. The area under ab represents work done during the expansion stroke. bc and da are isometric and involve no work. The com-

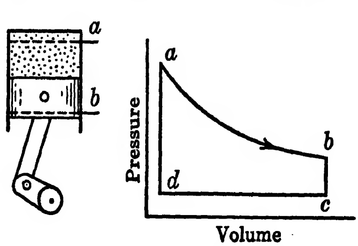


Fig. 4–6. Power cycle.

* Thermodynamics, Chapter 1.

pression stroke *cd* absorbs work so the net output of this power cycle is the difference between the areas under *ab* and *cd*, which is the same as the area *enclosed* by the outline of the complete cycle. Notice that the direction of this cycle has been *abcd;* that is, clockwise. Had it been *adcb* it would have been a compressor or heat pump cycle. But in any event, and for any other combination of processes yielding a closed cycle, the enclosed area is the mechanical work involved. If the pressure scale is 1 in. = P lbs. per sq. ft. and the volume scale 1 in. = V cu. ft., one square inch of cycle area represents PV ft. lbs. of work.

The P–V diagram is not as informative as plots on other planes in the case of "flow machines" such as turbines. Temperature-entropy (T–s) will often prove to be the better medium for graphical study of these prime movers. Under certain conditions (reversible expansions) the area enclosed by a T–s cycle is work involved, but in B.t.u. units. Were the scales of a T–s graph 1 in. = T deg. and 1 in. = s B.t.u. per deg., a unit area would again be the product of the scales, Ts B.t.u.

4–7. Carnot Cycle. An ideal cycle which has for a long time occupied a prominent place in thermodynamics is a reversible cycle, first presented in 1824 by Sadi Carnot. This cycle may be assumed to operate on one of the permanent gases, say air. It has four processes, two of them isothermal, two adiabatic. These processes are alternated on the cycle, there being both an isothermal and adiabatic phase of expansion and a repetition of the same during compression. This cycle is readily drawn on the T–s plane since isothermal and adiabatic (constant s) processes are straight lines normal respectively to the T and s axes.

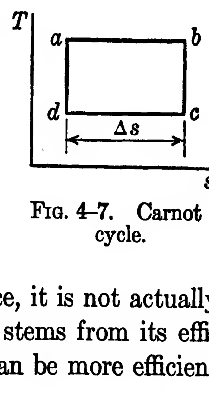
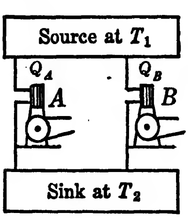


FIG. 4-7. Carnot cycle.

Although this cycle has been approximated in practice, it is not actually the basis of any existing prime mover. Its importance stems from its efficiency. It can be demonstrated * that no heat engine can be more efficient

* Engines **A** and **B** are taking heat from a source at T_1, converting the available fraction into work, and rejecting the unavailable to a sink at T_2. We postulate that (1) engine **B** is reversible, (2) engine **A** is more efficient than **B**, (3) the engine sizes are selected so that the power output is the same. Since **B** is the less efficient it will require more heat input than **A** for the same work output, therefore $Q_B > Q_A$.



Now let **B** be reversed and function as a heat pump, being driven by **A**. According to postulate (3) above, **A**'s output will just power **B** reversed. Q_A is taken from the source as before, but now Q_B is discharged into the source. Thus we have a self-contained system capable of transferring a net quantity of heat $Q_B - Q_A$ from a lower to a higher temperature. The reader is invited to discover which of the laws of thermodynamics this contradicts, and which of the three postulates above must therefore necessarily be erroneous.

than a reversible one; consequently, the Carnot cycle has as high a thermal efficiency as any. In addition to this, its thermal efficiency is very simply stated so it has become a convenient "yardstick" of comparison for all heat power cycles. The work per cycle is the area enclosed on the T–s plane. This area is $(T_1 - T_2)\Delta s$. The heat supplied to the cycle is that needed on the isothermal expansion ab. This is $T_1\Delta s$.* Thermal efficiency is the ratio of these. Let η_C be our Carnot cycle efficiency, then

$$\eta_C = \frac{T_1 - T_2}{T_1}.$$

Nothing could be simpler or more informative of what heat power cycles require for good performance than the above.

4-8. Energy of Jets. A jet is a rapidly moving fluid stream launched into space from a nozzle, orifice, or other mouthpiece. Bernoulli is credited with an important law governing the flow of liquids (i.e., incompressible fluids). Consider a liquid at various successive stations along its line of flow. At any station the physical elevation above some zero datum is y, the fluid unit pressure P, and the linear velocity v. The total "head" of the liquid at this point is:

$$\mathscr{H} = y + \frac{P}{\rho g} + \frac{v^2}{2g}.$$

ρ is the mass density of the liquid, g the acceleration of gravity. ρg is the specific weight. By use of appropriate units the head $\mathscr{H}$ may be rendered in feet of the liquid. Bernoulli's law states that if the flow takes place without external interference $\mathscr{H}$ is the same at all stations along the flow, although the three terms which compose it may vary from point to point.

Since $\mathscr{H}_1 = \mathscr{H}_2$,

$$y_1 + \frac{P_1}{\rho g} + \frac{v_1^2}{2g} = y_2 + \frac{P_2}{\rho g} + \frac{v_2^2}{2g}.$$

If stations one and two are near each other, as for a nozzle, it is usually considered that $y_1 = y_2$. When the discharge is to the atmosphere, $P_1 - P_2 =$ gage pressure P.

Then from the above,

$$\frac{P}{\rho g} = \frac{v_2^2 - v_1^2}{2g}.$$

As it is the velocity-producing quantity, $P/\rho g$ will be designated *velocity head* and represented by H. When water flows through a nozzle to form a jet, it is usually considered that v_1^2 is negligible compared to v_2^2, so Bernoulli's law

* $T\Delta s$ is *heat* in accordance with ideas set forth in **Entropy**, Chapter 2.

is reduced to the following expressions for the flow of an incompressible fluid from a nozzle:

$$H = \frac{v_2^2}{2g}; \quad \text{or} \quad v_2 = \sqrt{2gH}.$$

Thus the water jet will attain a theoretical velocity of $\sqrt{2gH}$ ft. per sec. when it is produced by an effective pressure head of H ft. of water. The actual jet velocity is 1 or 2% less than the theoretical.

A water jet of A sq. ft. cross-sectional area moving at v ft. per sec. contains energy at a rate equivalent to $\frac{1}{566} Av^3$ hp. To obtain this power from the jet a machine would have to bring the water to rest without friction or turbulence. Impulse hydraulic turbines recover about 70% of the available power by deflecting the jet with blades or buckets mounted on the rim of a rotating disk.

The force acting on a blade which deviates a jet of water through a velocity change of Δv is $\frac{62.4\Delta v}{g}$ lbs. per cu. ft. of water per sec. flowing. Δv may be a change in magnitude, a change in direction, or both.

As gas is a compressible fluid, the velocity attained in a jet cannot be evaluated from Bernoulli's principle. If one uses adiabatic expansion from P_1 to P_2 lbs. per sq. ft. in a properly shaped nozzle, and assumes the velocity of approach to the nozzle zero and that there is no chemical change during expansion, the jet velocity is:

$$v = 8\sqrt{\frac{RT_1}{z}\left[1 - \left(\frac{P_2}{P_1}\right)^z\right]} \text{ ft. per sec.}$$

R and z are characteristics of the gas, R being a common gas constant (i.e., *53.4* ft. per deg. for air), and z being $(c_p - c_v)/c_p$, wherein the c's are specific heats, B.t.u. per lb. per deg., at constant pressure and constant volume.

$$z = .286 \text{ for air under } 1000° \text{ F.}$$

$$z = .23 \text{ to } .30 \text{ for products of combustion of fuels.}$$

T_1 is the absolute temperature at pressure P_1.

If reliable specific heat data are available, it may be noted that the change of enthalpy Δh of the gas flowing through the ideal nozzle is $c_p\Delta T$, where ΔT is $T_1 - T_2$. Then the analysis of velocity by the energy method given for steam nozzles in the following paragraphs may be employed for a gas jet.

Although gas jets have not been in common use heretofore, rocket propulsion and other uses of gas jets may serve to focus more attention on this subject.

A jet of steam is produced by the expansion of steam from a higher to a lower pressure. Nearly always the character of expansion is a close approximation of the adiabatic at constant entropy since nozzles neither impart nor remove heat from the steam flowing through them. The nozzle may be carefully shaped, based on the theory of compressible flows through nozzles, or it may be a crude nozzle such as a small bored hole in a plate. The less perfect the nozzle, the lower the steam jet velocity will be and the more uncertain

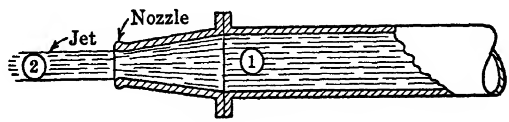


FIG. 4–8A. Water jet.

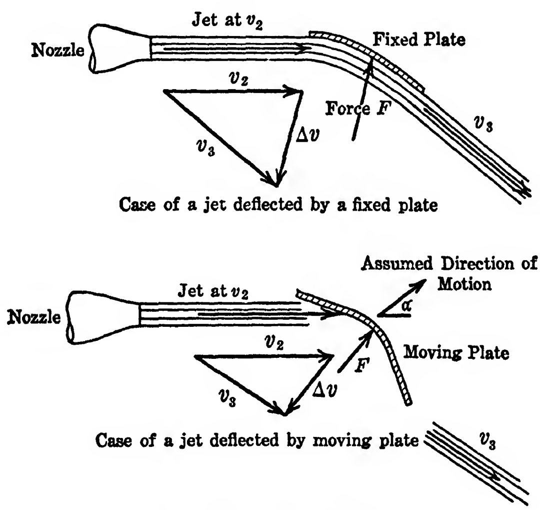


FIG. 4–8B. Deflected jets.

the jet direction. Steam, like gas, is so light in weight per unit volume and its jet velocity usually so large that the effect of gravity on the path of the jet is negligible so the axis of the nozzle determines the position of the jet for a considerable distance from the nozzle mouth.

The ideal velocity of a steam jet resulting from an adiabatic expansion may be calculated from the principle of conservation of energy. If one lb. of steam in expanding through a nozzle (which it approached at a velocity of v_1) suffers a change of enthalpy of Δh B.t.u., and if conditions of expansion are ideal, namely no friction, turbulence, over-or-under-expansion, or supersaturation, v_2^2 will be:

$$v_2^2 = 224^2 \Delta h + v_1^2.$$

This is derived by equating the loss of enthalpy to a gain in kinetic energy. Actual jet velocity approaches closely the ideal.

Although of low density, steam jets may represent great power because of extremely high velocities which may be produced with moderate pressure drops. The power of steam jets is put to use in numerous ways. Steam turbines are machines for producing jet power, then absorbing it on blades which in turn produce power as a rotating torque at a shaft to which the blades are indirectly attached. Certain refrigerating apparatus uses steam jets for compression, as do also ejectors for evacuating air from regions of high vacuum. Steam jets are the source of power for injector pumps and are often used for blowing and cleaning operations.

The kinetic energy represented in a pound of fluid moving at v ft. per sec. is $v^2/2g$ ft. lbs. The power in a jet of w lbs. per sec. flow is $wv^2/2g$ ft. lbs. per sec., or $wv^2/(2 \times 550g)$ horsepower. If a jet of w lbs. per sec. flow strikes and is deflected by a stationary plate, the plate will receive a thrust of $w\Delta v/g$ lbs. from the jet, but no work will be done on it since it is at rest. Δv is obtained by vectorial subtraction since velocity has direction as well as magnitude. Thrust F will be directed parallel to Δv. The jet will have the same energy at v_3 that it possesses at v_2. However, if the plate can move, as in Figure 4–8B, some velocity v_3 will result from the combination of that motion and the plate curvature. As in case A, $F = w\Delta v/g$ in the direction of Δv. The jet energy at v_3 is less because the moving plate received work due to the product of its motion and the component of F in the direction of plate motion.

Example 1: A jet of water issuing from a $\frac{1}{2}$-in. diameter nozzle attached to a pipe in which the water pressure is 50 psi. gage, strikes a stationary curved plate and is deflected, without turbulence, through an angle of 30°. Calculate the jet velocity, power, and plate pressure, neglecting the effects of friction in the nozzle and on the plate.

The area of the nozzle opening is $\frac{1}{2}^2\pi/(4 \times 144)$ or .001365 sq. ft. The pressure is converted to equivalent static head of water, using the relation $H = P/\rho g$. P must be in lbs. per sq. ft. ρg is 62.4 lbs. per cu. ft. for water. Hence,

$$H = \frac{50 \times 144}{62.4} = 115 \text{ ft.}$$

$$\text{Jet velocity } v = \sqrt{2 \times 32.2 \times 115} = 86 \text{ ft. per sec.}$$

$$\text{Jet power} = \frac{.001365 \times 86^3}{566} = 1.54 \text{ hp.}$$

Change of velocity effected by the plate is twice $v \sin 30°/2$.

$$\Delta v = 2 \times 86 \times .2588 = 44.5 \text{ ft. per sec.}$$

$$\text{Force against the plate} = \frac{62.4 \times 44.5}{32.2} = 86.2 \text{ lbs. per cu. ft. of water per sec.}$$

Since the water flow is Av or $.001365 \times 86$ cu. ft. per sec., the total force $F = 86.2 \times .001365 \times 86 = 10.1$ lbs.

Example 2: The nozzles in a certain gas turbine are designed for efficient expansion of the gas from 1000° and 55 psi. abs. to atmospheric pressure. Find the jet speed leaving the nozzle. As combustion in a gas turbine under these conditions represents use of an overwhelming amount of excess air, the gas constants will be taken as for air, i.e., $R = 53.4$, $z = .286$. Substituting in the equation given above,

$$v = 8\sqrt{\frac{53.4(1000 + 460)}{.286}\left[1 - \left(\frac{14.7}{55}\right)^{.286}\right]} \text{ ft. per sec.}$$

A simple way of solving $(.267)^{.286}$ without using negative logarithms is to write it this way:

$$(.267)^{.286} = \left(\frac{1}{\frac{1}{.267}}\right)^{.286} = \frac{1}{(3.74)^{.286}} = \frac{1}{1.46} = .685.$$

$$v = 8\sqrt{273{,}000(1 - .685)} = 2345 \text{ ft. per sec.}$$

An alternate solution, assuming $c_p = .245$ B.t.u. per lb. per deg., would use the equation $v_2 = 224\sqrt{\Delta h}$ since v_1 can be assumed negligibly small. But we shall have to know T_2 as well as T_1. Since $P_1V_1^\gamma = P_2V_2^\gamma$ in nozzle flow, and $P_1V_1/T_1 = P_2V_2/T_2$, the volume terms may be algebraically eliminated between these two equations, yielding $T_1/T_2 = (P_1/P_2)^{(\gamma-1)/\gamma}$.

The exponent is identical with the z term of this section, hence

$$\frac{(1000 + 460)}{T_2} = \left(\frac{55}{14.7}\right)^{.286} = 1.46; \quad T_2 = 1000° \text{ R.}$$

$$v = 224\sqrt{c_p\,\Delta T} = 224\sqrt{.245(1460 - 1000)} = 2370 \text{ ft. per sec.}$$

Example 3: A jet of steam is produced by adiabatic release of 85 B.t.u. per lb. during expansion through a nozzle. This jet then strikes a curved moving blade whose direction of motion makes an angle $\propto$ with the steam jet (in the general direction shown in Figure 4–8B). The blade speed and its curvature is such that the steam leaves at right angles to its original direction with a velocity of one-third the nozzle velocity. What are the steam speeds, the blade force, and the energy delivered per lb. steam per sec.?

$$v_2 = 224\sqrt{85} = 2065 \text{ ft. per sec. } (v_1 \text{ assumed } 0).$$

$$v_3 = \tfrac{1}{3}(2065) = 688 \text{ ft. per sec.}$$

The vector triangle of velocities is a right triangle with $\Delta v = \sqrt{2065^2 + 688^2} =$ 2175 ft. per sec.

Blade force is 2175/32.2 = 67.5 lbs. per lb. steam per sec. directed parallel to Δv.

Energy delivered to the blade = kinetic energy of the v_2 jet minus kinetic energy of the v_3 jet (assuming frictionless streamline flow).

$$\text{Energy delivered} = \frac{2065^2}{2 \times 32.2} - \frac{688^2}{2 \times 32.2} = 58{,}800 \text{ ft. lbs. per lb. steam per sec.}$$

4–9. Electrical Energy. Considerable data on transformation ratios involving electrical energy have already been introduced. Using these data, some

typical conversions are investigated. The electric motor which receives a flow of amperes under a pressure of volts can convert most of the watts of input into output power. Motor efficiencies are typically in the range 75% to 95%. The losses appear as heat at the bearings, in the conductors, and in the magnetic iron core. Consideration of the action whereby an electron flow in conductors goes over into mechanical work (or heat energy) or vice versa, is outside the contemplated scope of this book. However, the reader may well acquire an elementary knowledge of overall energy transformations without, in this case, obtaining an exact picture of how it is accomplished. The power output of a motor being P horsepower, and its overall efficiency η, the necessary electrical input is $.746P/\eta$ kilowatts. The constant .746, being kilowatts per horsepower, is established dimensionally thus:

$$\frac{\dfrac{\text{ft. lbs. per min.}}{\text{horsepower}}}{\dfrac{\text{ft. lbs. per sec.}}{\text{watt}} \times \dfrac{\text{sec.}}{\text{min.}} \times \dfrac{\text{watts}}{\text{kilowatt}}} = \frac{\text{kilowatts}}{\text{horsepower}},$$

or,

$$\frac{33{,}000}{.7375 \times 60 \times 1000} = .746 \text{ kw. per hp.}$$

In a direct current circuit the relation between I amperes, E volts, and kilowatts is, of course, $EI/1000 = \text{kw.}$ Consequently, the motor which delivers P horsepower will draw from a circuit operated at E voltage a current I in accordance with the following equation:

$$EI = 746\frac{P}{\eta}.$$

Power in alternating current circuits is influenced by two additional electrical factors: the number of phases and the power factor. Usually motors are either single-phase or three-phase. Single-phase motors have two electrical leads connecting them to the source of energy—three-phase motors have three leads. The power factor term accounts for the decrease of power in the circuit when the current and voltage do not alternate exactly in phase with each other. Let f be the power factor. The power equation for single-phase motors is:

$$EI = 746\frac{P}{f\eta},$$

and for three-phase motors:

$$EI = 746\frac{P}{\sqrt{3}f\eta}.$$

Example 1: A single-phase motor operating on 110 volts with 80% power factor is to deliver $\frac{1}{2}$ horsepower at full load. The efficiency is estimated to be 82%. How much current will be drawn from the power supply?

Using the second of the foregoing power equations:

$$110\,I = \frac{746 \times \frac{1}{2}}{.80 \times .82},$$

$$I = 5.2 \text{ amperes.}$$

Example 2: What is the efficiency of a direct current motor which can overcome a torque of 8.75 pound-feet at 1200 rpm. while receiving 7.5 amperes at 220 volts?

A torque of 8.5 lb. ft. revolved once (2π radians) represents $8.5 \times 2\pi$ ft. lbs. mechanical work. When this happens 1200 times a minute, the power is $8.5 \times 2\pi \times 1200/33{,}000$ hp. Substituting this for P in the first power equation,

$$7.5 \times 220 = 746\,\frac{\left(\dfrac{8.5 \times 2\pi \times 1200}{33{,}000}\right)}{\eta},$$

$$\eta = 87.7\%$$

The power equations for generators are similar but the efficiency factor must be moved from the denominator to the numerator of the fraction which includes it. For a generator, P becomes the necessary input power and E and I the electrical output. The efficiency factor η must include the losses in any external control resistors, also the energy necessary to drive the exciter * of an alternating current generator.

Electricity is employed for the generation of heat in industrial, commercial, and domestic use. When compared with direct heating from fuels, electric heating will show a much higher cost, since the equipment for initially converting heat energy into electrical energy fails to convert, at its best, some 70% of the heat units. Nevertheless, since electrical heating apparatus produces the heat at the very spot desired, and exactly in the desired amount, the difference in cost is not so pronounced, because of the economy of electric heating apparatus. For localized heating, such as cooking, soldering and welding, electric heating is especially good; also it is used where cost is not an important item, viz., apartments, shipboard, hospitals, etc.

Heat may be obtained from electrical energy by means of resistance heating, induction heating, and the arc. Resistance heating is used in most domestic appliances, industrial ovens, and the like. Arcs are used for welding, lighting, and furnaces. Induction heating has been employed in welders, furnaces, and therapeutic devices.

The amount of heat developed in resistance heating is proportional to the resistance of the heating element, and to the square of the current flowing.

* A small direct current generator used to supply field current to the alternating current machine. It is generally direct-connected to the shaft of the large machine.

The electrical energy converted into heat when I amperes flow through a resistance of R ohms for t seconds is $I^2 \times R \times t$ watt seconds. There are 1055 watt sec. in a British Thermal Unit.

Example 3: Two coils are connected across a 110-volt circuit. One has a resistance of 20 ohms, the other 30 ohms. What heat is evolved from these coils per minute?

The current flow through the coils is 110/20 and 110/30 amperes, respectively. The watt seconds heat energy released per minute are:

$$I^2Rt = (\tfrac{110}{20})^2 \times 20 \times 60 + (\tfrac{110}{30})^2 \times 30 \times 60 = 8{,}750{,}000 \text{ watt sec.}$$

$$\text{Heat equivalent} = \frac{8{,}750{,}000}{1055} = 8280 \text{ B.t.u.}$$

Example 4: An electric water heater is wanted which will heat 2 lbs. of water from 60° to 212° in 10 min. The heating element is a single coil across a 110-volt circuit. What working resistance should it possess, and how many amperes must flow?

The water heating will require 2(212 − 60), or 304 B.t.u. In this case,

$$I^2Rt = 304 \times 1055.$$

Using Ohm's law, $I = E/R$, and noting that $t = 600$ sec.,

$$\left(\frac{E}{R}\right)^2 R \times 600 = 304 \times 1055.$$

$$\frac{110^2}{R} = 534, \quad R = 22.7 \text{ ohms.}$$

$$I = \frac{110}{22.7} = 4.84 \text{ amperes.}$$

4–10. Release of Atomic Energy. The story of atomic energy goes for its beginnings to the discovery, in 1896, of the phenomenon of radioactivity. So much has since been disclosed through theorization and experiment, and so many gifted scientists are to be credited with advancing man's knowledge of atomic structure, and in particular the existence and release of the enormous quantity of energy contained there, that it is quite impossible to relate that story in these pages.* A brief account can be rendered of the processes leading to unlocking the energy of the atom for the purposes of warfare. The highly explosive character of this achievement renders it of little use in peacetime, but there is no reason to believe that it is impossible to tame the behemoth and use it to attain a happier end than that for which it was so frantically sought.

In Chapter 1 the structure of matter was pictured in some detail. Stable and unstable nuclei were mentioned. It may be assumed that the disintegra-

* The first comprehensive historical account of radioactivity with reference to the practical release of atomic energy is believed to be the authorized U. S. Government report *Atomic Energy (for Military Purposes)* by H. D. Smyth. Princeton University Press.

tion of any stable system requires work to be done on it. Energy must be supplied to break up an atom having a stable nucleus. Since mass and energy are equivalent, the total mass of the stable nucleus should be exceeded by the total mass of the separate neutrons and protons that compose it. The difference is the "binding energy" of the nucleus. The alpha particle is the nucleus of a helium atom (see Figure 1–1). This nuclear mass is actually 4.00280. However, the mass of 2 protons + 2 neutrons * is $2 \times 1.00758 + 2 \times 1.00893$, or 4.03302. Here is a difference of .03022, which necessarily must be the nuclear binding energy. As we have previously seen, a gram of matter is equivalent to 66.3×10^{12} ft. lbs. of energy. A pound, then, is $30{,}150 \times 10^{12}$ ft. lbs., or 38.7×10^{12} B.t.u. In the case of one helium atom, there is a weight of $.03022 \times \dfrac{3.69 \times 10^{-27} \, *}{1.00758} = .1107 \times 10^{-27}$ lb. in nuclear binding energy. This becomes $38.7 \times 10^{12} \times .1107 \times 10^{-27} = 4.28 \times 10^{-15}$ B.t.u. per helium molecule. There are 2748.8×10^{23} molecules in a pound molecular weight of any gas.† Helium has an atomic weight of 4 and is a monatomic molecule. Therefore the number of molecules, and hence also the nuclei, per pound, is of the order of 687.2×10^{23}. The nuclear binding energy is, therefore, $687.2 \times 10^{23} \times 4.28 \times 10^{-15} = 2.95 \times 10^{12}$ B.t.u. per lb. This is an enormous quantity of energy, judged by ordinary industrial standards, and is most interesting, for if free protons and neutrons could be assembled into helium nuclei, this energy would be released. If only one part in a million of these actually combined into nuclear particles, the mass would rise to temperatures of over a million degrees by absorption of the released energy. It represents thermodynamic potentials enormously above any that can be secured by combustion. Evidently it was worth while to explore the field of nuclear energy.

It was found in 1939 that a free neutron striking a Uranium nucleus was sometimes absorbed, causing the nucleus to split into two fragments, releasing a large quantity of energy. This process has come to be called nuclear "fission." Uranium is radioactive and is found in nature in a mixture of two isotopes, 99.3% U^{238} and .7% U^{235}. The final products of Uranium fission are an atom each of Barium and Krypton, one or more free neutrons, and energy. Actually the fission produces two unequal fragments of mass numbers of about 90 and 140. There is subsequent radioactive decay by the emission of electrons until stable nuclei are formed. The free neutrons may proceed to cause other fission, creating a chain reaction, or the action may cease.

Thus, prior to the intensified search for atomic energy in wartime, it was

* Footnote page 2.

† This is the *Avogadro Number* in the English system. A mass of experimental evidence supports the hypothesis of Avogadro and places the number of molecules in a gas at 6.06×10^{23} molecules per gram molecular weight.

known that (1) fission was possible, (2) radioactive elements could be artificially produced, and (3) there was a possibility of obtaining a neutron chain reaction which might be self-supporting.

Thermal (slow) neutrons were found to fission U^{235}, but not U^{238}. The high-speed neutrons emitted by the fission needed to be slowed down in order to carry on the chain reactions since high-speed neutrons had little tendency to collide with nuclei and, when they did collide, fission did not necessarily ensue. In passing through a "moderator" material (stable atoms of low atomic weight; graphite, heavy water, Beryllium) the neutrons are slowed. Large numbers of these slow neutrons in a mass of ${}_{92}U^{238}$ * would be absorbed, creating ${}_{92}U^{239}$, each atom of which immediately emits an electron leaving a new element, Neptunium, ${}_{93}Np^{239}$. Neptunium quickly decays into a new element, Plutonium, Pu, which is found to be relatively stable, although radioactive. In terms of nuclear notation this is written: †

$$ {}_{92}U^{238} + {}_{0}n^{1} \rightarrow {}_{92}U^{239} \rightarrow {}_{93}Np^{239} + {}_{-1}e^{0} $$

$$ {}_{93}Np^{239} \rightarrow {}_{94}Pu^{239} + {}_{-1}e^{0}. $$

Apparently Plutonium was sought for the purposes of atomic bomb manufacture. It could be chemically separated from the untransmuted Uranium and enormous plants were set up to manufacture it. The details of the use of Pu^{239} and U^{235} in a fission bomb have been withheld for security reasons. The process is not important to heat power study since it is explosive. Incidental energy produced during Plutonium manufacture was wasted in cooling water.

Of more interest to the future of atomic energy in peaceful applications is the recovery of the energy of fission of U^{235} or some other fissionable element. The principle of operation of an atomic bomb or power plant utilizing Uranium fission is simple enough. If one neutron causes a fission that produces more than one new neutron, the number of fissions may increase tremendously with the release of enormous amounts of energy. It is a question of probabilities. Neutrons produced in the fission process may escape entirely from the Uranium, may be captured by Uranium in a process not resulting in fission, or may be captured by an impurity. Thus the question of whether a chain reaction does or does not go depends on the result of a competition among four processes:

1. Escape.
2. Non-fission capture by Uranium.
3. Non-fission capture by impurities.
4. Fission capture.

* Signifies Uranium with mass 238 and nuclear charge 92.

† ${}_{0}n^{1}$ represents a neutron, ${}_{-1}e^{0}$ an electron, or beta particle.

If the loss of neutrons by the first three processes is less than the surplus produced by the fourth, the chain reaction occurs; otherwise it does not. If such a reaction is going to be of use, we must be able to control it. The problem of control is different depending on whether we are interested in steady production of power or in an explosion. In general, the steady production of atomic power requires a slow-neutron-induced fission chain reaction occurring in a mixture or lattice of Uranium and moderator, while an atomic bomb requires a fast-neutron-induced fission chain reaction in U^{235} or Pu^{239}.

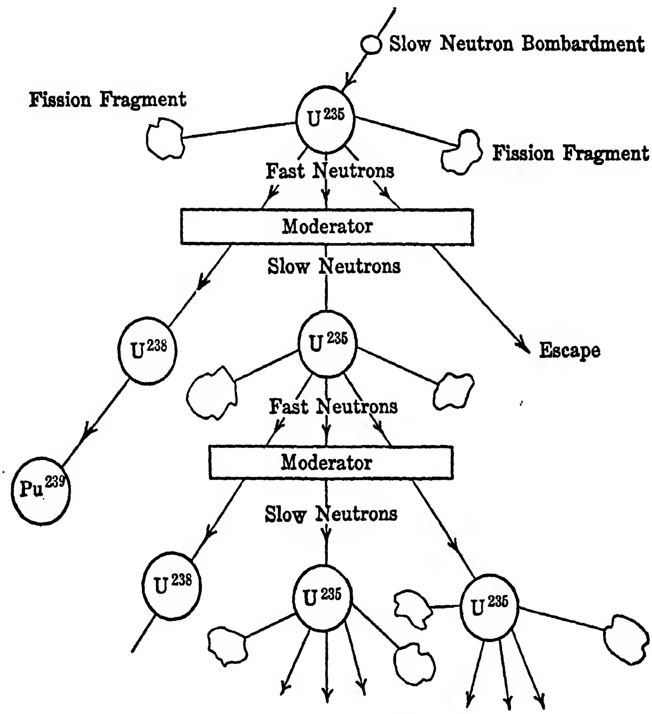


FIG. 4–9. Chain reaction.

The U^{235} chain reaction could, for atomic power purposes, be carried out without the presence of U^{238}. The cost of isotopic separation is very high, but some improvement of the 0.7% natural content of U^{235} in Uranium may be had by diffusion, centrifuge, or electromagnetic systems of separation. Probably in such a power plant Plutonium would be a by-product. The chain reaction could go as shown in Figure 4–9. Basing ideas on such details as have been released on Plutonium production, heat might be generated in a medium (liquid, vapor, or gas) by using that medium to cool the "piles" which are the assemblies of Uranium—moderator mixtures in the plants producing the chain reaction. The heated medium could then be employed as industrial heat—or to drive steam or gas turbines. The Uranium, sealed

in aluminum cans, is placed in the pile where the chain reaction is controlled by imposing variable barriers in the path of the free neutrons. The piles using natural Uranium will not maintain the chain reaction unless many tons of the

Fig. 4-10. Concentrated energy. "Baker Day" underwater atomic bomb explosion, July 25, 1946. (Photo by Army-Navy Joint Task Force 1.)

material are present. The "critical mass" at which the reaction will maintain itself decreases as the percentage of U^{235} is artificially increased in the Uranium, so permitting more compact energy sources. Whether this would justify the cost of enrichment is, again, a matter associated with future commercial attempts to tap this source of energy.

PROBLEMS

1. Rule a large-sized table similar to the one shown here, and fill in as many of the blank spaces as you can by thinking of practical examples of energy in action.

	Heat	Work	Electricity	Radiation
Heat				
Work			Generator	
Electricity		Motor		
Radiation				

The blocks in the diagonal line will be cases of energy transfer; all others are energy transformation.

2. A certain airliner weighs 36,500 lbs. when at rest on the airport. What is its equivalent weight when in flight at 350 mi. per hr.? Use data from Chapter 1. Why is this difference in weight not ordinarily taken into consideration?

3. A heat engine received heat at the rate of 5150 B.t.u. per min. and converted 10% of it into mechanical work available at the engine shaft. What was the horsepower thus made available?

4. A pound of dry saturated steam at 100 psi. abs. has 20% of its enthalpy (above 32° F) converted into mechanical work. How many foot-pounds appeared?

5. Twenty-five per cent of the heat energy (above 32° F) contained by a pound of steam entering a nozzle is converted into kinetic energy of the issuing jet. The steam was 98% dry at 50 psi. abs. How many B.t.u. of heat energy remain in the steam? What is the speed of the jet, ft. per sec.?

6. A hoisting drum with friction brake can lower a 1500-lb. weight at a uniform speed of 10 ft. per sec. What heat is developed at the hoist brake per minute?

7. Results of a test on a 5-hp. gasoline engine showed 26% of the heat of the fuel going into useful output, 30% into hot exhaust gas, 32% into cooling water. How many B.t.u. were lost from friction and radiation, per minute?

8. A nozzle produces a water jet having a diameter of 2 in. and a velocity of 200 ft. per sec. Density of water 62.5 lbs. per cu. ft. Find the kinetic energy per sec. in this flow. Were 92% of this energy converted into electrical form by a hydraulic turbine and generator, how many kilowatts would be produced?

9. A rifle bullet weighing 1 oz. travelling at 1200 ft. per sec. strikes and is embedded in a stationary target weighing 2 lbs. Specific heat of target material .5 B.t.u. per deg. per lb., of bullet, .2 B.t.u. per deg. per lb. Prior to contact temperatures were: target, 60° F, bullet 180° F. What was the equilibrium temperature after impact? Discount any heat losses to the surroundings prior to attainment of equilibrium temperature.

10. A set of transmission gears receives a torque of 105 lb. ft. at 2550 rpm. The delivered torque is 400 lb.-ft. at 600 rpm. Find the mechanical efficiency of the transmission. Why do the losses appear as heat? How many B.t.u. per minute appear? Note: Torque in lb.-ft. $\times$ rotation, in radians, = work in ft.-lb.

11. An electron-volt is equivalent to 1.591×10^{-19} Joule. Show that a Mev is 4.45×10^{-20} kw. hr.

12. Thirty-five per cent of the heat in the fuel for an internal combustion engine is transferred to the cooling water which, in turn, releases it to air drawn through the cooling "radiator." Twenty-five per cent of the heat of the fuel is converted into an output of 56 hp. Air is heated 42° by the radiator. How much air (pounds per minute) is flowing through the radiator?

13. What is (a) the arithmetical and (b) the true thermal mean temperature difference of the heat transfer described in Example 3, page 83?

14. Heat is transferred from condensing steam to water flowing inside tubes with an overall conductance (U) of 500 B.t.u. per hr. per sq. ft. per deg. Steam temperature 100°, water in 70°, water out 90° F. (a) Calculate the true m.t.d. (b) What is the rate of heat transfer per minute through 400 one-inch diameter tubes 10 ft. long?

15. A 20-lb. chunk of steel is warmed to 200° F in an oven, then transferred to a 10-gallon can half full of water at 60° F. How much heat was transferred from the oven to the water? Is the mode conduction, convection, or radiation? Can you make a statement about the quantity of heat leaving the oven in the act of withdrawing the steel?

16. A roof is composed of 1 in. of pine wood covered by sheet steel. Assume infinite conductance for the covering. Also consider the air film resistances negligibly small. Inside temperature 70°, outside 0° F. By what percentage is the heat loss reduced by insulating the roof with a 2-in. thick layer of rock wool?

17. Assume that the wall of Figure 4–2b is made of 4 in. of red brick, 2 in. of cork, and $\frac{1}{2}$ in. plaster. Air film conductance 1.5 B.t.u. per hr. per sq. ft. per deg. F. Estimate the heat leakage through a wall 10 ft. long by 8 ft. high. Room temperature 70° F, refrigerator temperature 18° F.

18. A concrete wall 12 in. thick has its surface temperatures measured. The surface on the side next to 50° F ambient atmospheric temperature is found to be at 54° F. That next to an ambient temperature of 70° F is found to be 65° F. (a) Calculate the heat flow through the wall. (b) Estimate the air film conductances, using the known heat flow.

19. An electrically heated oven with 25 sq. ft. surface exposure has 6-in. thick walls of a special insulating refractory material. The outside wall temperature is 150° F when it is 1200° inside. To maintain this temperature the heaters use 3 kw. What is the conductivity of the wall material?

20. Data of Example 2, page 82. Determine the temperatures at the layer surfaces and plot a temperature variation chart to scales as follows: 1 in. = 20° F, 1 in. = 2 in. of wall thickness. (Method: ΔT for warm air film is calculated thus. $Q = 12 = 1.8 \times 1 \times \Delta T$, $\Delta T = 6.7°$; therefore, warm wall temperature $= 80 - 6.7 = 73.3°$ F.)

21. During the stroke of the piston of a certain engine, combustion of fuel added 140 B.t.u., cylinder walls absorbed 13 B.t.u., mechanical friction accounted for 120 ft. lbs., and there were 82 more B.t.u. in the products of combustion than the air originally possessed. How many foot-pounds of useful work were delivered by the piston?

22. A cylinder 4 in. in diameter is closed at one end by a movable piston. It contains gas under pressure. The piston moves outward 18 in. against average resistance of 120 psi. in an *adiabatic* expansion. How much energy must be supplied from the internal energy of the gas? How much will this cool the gas if there is .03 lb. of it in the cylinder and its specific heat is .24 B.t.u. per deg. per lb. at constant pressure, .17 B.t.u. at constant volume. $R = 50$.

23. What is the maximum availability of heat for doing work when that heat is obtained from a region at 1800° F, and unavailable heat is finally rejected to a region of 60° F?

24. Were the available energy of a pound of steam 155 B.t.u., how many pounds of steam would be used by an ideal engine for each horsepower hour of work it produces?

25. The expansion stroke of the cycle of Figure 4–6 follows the law $PV^2 = C$. At a the pressure is 250 psi., the volume .2 cu. ft. The volume at b is .6 cu. ft., pressure at d is 14.7 psi. Draw the cycle, and find its area in foot-pounds. Scales 1 in. = 50 psi., 1 in. = .1 cu. ft.

26. A refrigerating plant is withdrawing 88 B.t.u. per min. from a cooler and discharging it to the atmosphere. It is driven by a $\frac{1}{2}$-hp. motor. How many B.t.u. per minute are discharged from this heat pump system?

27. An ideal cycle working as a power cycle makes the fraction $(T_1 - T_2)/T_1$ of the heat supplied it available for work. When reversed and acting as a heat pump it is called on to extract 450 B.t.u. per min. from an 18° F space and discharge it at 75° F. What hp. input is required?

28. It is desired to plot the P–V diagram of an air compressor which has an 11 in. bore and a 12 in. stroke to scales such that a square inch represents about 100 ft. lbs. The discharge pressure is 50 psi. What scales would you recommend for a well-balanced diagram?

29. A steam engine has been equipped with a special instrument to determine its P–V diagram. A diagram is produced that is 3.2 in. long, this representing a piston displacement of 610 cu. in. The 2.7-in. height of the diagram represents a pressure of 162 psi. What does a sq. in. of area represent?

30. On a T–s plane (1 in. = 100° Rankine, 1 in. = .3 entropy) plot a line representing the conversion of 1 lb. water originally at 32° F into dry saturated steam at 50 psi. abs. Measure the area under this plot in sq. in. and interpret it in B.t.u.'s. Compare result with h_g at 50 psi. in the saturated steam tables. The "T" scale must be in deg. Rankine. Why?

31. Using scales of 1 in. = 100° Rankine and 1 in. = .3 entropy units, plot a graph representing the conversion of water originally at 32° F into superheated steam at 40 psi. and 400° F. Measure the area under this plot in sq. in. and interpret it in B.t.u.'s. Compare your result with enthalpy as shown in the steam table for this superheated condition.

32. What is the best thermal efficiency that could be obtained by any cycle working between a 150-psi. steam boiler (no superheater) as the source of heat and the atmosphere (60°) as the sink?

33. A Carnot cycle, which receives heat at 1500° F, discharges 15 B.t.u. of unavailable heat to a region at 40° F each cycle. What is its efficiency? How many ft. lbs. of work are done per cycle?

34. The inventor of a certain heat power cycle claims an efficiency for it of 66% when the combustion chamber temperature is maintained at 2000° F and the exhaust is at 450° F. Submit a detailed proof (based on the second law) to prove that an impossibly high efficiency is claimed.

35. A water jet produced by a head of 100 ft. issues from a nozzle with 98% of its ideal velocity. Find the horsepower represented in a flow of 200 gal. per min.

36. What is the velocity of a jet of liquid of 91% specific gravity as it issues from a nozzle back of which there is a pressure of 25 psi. gage?

37. Calculate the velocity of an air jet produced by the adiabatic expansion of air from a region at 25 psi. gage to atmospheric pressure. Initial temperature 120° F.

Also, find the final temperature of the jet. Would this be the temperature registered on a stationary thermometer or on one moving with the jet?

38. Products of combustion of a gasoline engine are sent at 20 psi. engine exhaust pressure into the nozzles of a turbo supercharger. Engine exhaust temperature 800° F. Nozzles discharge the jet into atmospheric pressure. What is the jet speed? Use $R = 53.8$, $z = .26$.

39. Steam expands through a nozzle from 120 psi. dry and saturated state to 40 psi. Using the Mollier chart, trace the isentropic expansion and find Δh. What ideal steam velocity does this produce?

40. The power in a fluid jet is one-half of the rate of mass flow multiplied by the square of the velocity. Beginning with this item, derive the constant 566 of the equation for horsepower of a water jet.

41. What heat release by expansion is necessary to increase the speed of steam from 450 ft. per sec. to 1200 ft. per sec.? The steam in the jet is to be dry and saturated at 15 psi. Use the Mollier chart to discover what initial pressure and degree of superheat must be used to obtain the required Δh.

42. In an ideal steam nozzle the enthalpy drop increases the jet kinetic energy from $\frac{1}{2}mv_1^2$ to $\frac{1}{2}mv_2^2$. Use this fact to derive the constant 224 of the jet velocity equation.

43. A direct current motor converted 700 watts of electrical input into work of 420 ft. lbs. per sec. What was its efficiency (of energy conversion)?

44. A steam engine is belted to a direct current generator whose efficiency is estimated to be 88%. It is guessed that the belt is capable of transmitting 98.5% of the engine output to the generator. When the engine is delivering 25 hp., how many amperes are flowing at the generator terminals? Voltage 220.

45. Simple, single-phase A-C motors usually have a power factor of about 80%. Assume that the motor efficiency is 90% and estimate the current to a 440-volt motor when delivering 15 hp.

46. A three-phase A-C motor with 80% power factor, 90% efficiency (energy) is direct-connected to a fan. Instruments show that it is taking 18 amperes per phase from 440-volt lines. Estimate the power required by the fan.

47. In a certain industrial process 185 lbs. of a liquid (spec. heat .85 B.t.u. per deg. per lb.) are required to be heated 65° F in a 42-min. period. If an immersed electric coil is used to accomplish this, how many watts will it draw while in operation, and what must be its working resistance? Voltage 110.

48. Steam has been used for heating in an industry. In one process, 10 lbs. of 30 psi. abs. steam are condensed each 15 min. How many kw. of electric power would be required for the same job? How many 12-ohm resistors in parallel will produce this heating? Voltage 110.

49. An electric heater is wanted to heat 24 lbs. fuel oil per min. from 40° F to 150° F, prior to use in an atomizing burner. Estimate specific heat from Table 2–1. What wattage must the heater have to accomplish this?

50. A deuteron is an atomic particle composed of a neutron and proton held together by nuclear binding energy. On the atomic mass scale, the deuteron mass is 2.0146. Find the binding energy in Mev per deuteron.

51. Assume that all the energy released in the fission of U^{235} in an ounce of normal uranium is captured for use. Also assume that the final products are Barium (atomic weight 137.36), Krypton (atomic weight 83.7), the U^{238}, and energy. How many kilowatt hours are obtained per ounce?

52. Lithium, No. 3 in the periodic table, has an atomic weight of 6.940. Construct a hypothetical diagram of this atom for each of the isotopes mentioned on page 5. In what proportions are the two isotopes in normal Lithium?

CHAPTER 5

Mechanical Elements of Heat Engines

5-1. General. Here we are to consider some things that many of the different heat engines and heat pumps have in common. Principally this will serve to acquaint the reader with the mechanical elements of displacement equipment; however, occasion is taken to introduce some other matters of common import. A convenient classification for this purpose is to separate the equipment into *displacement* and *steady-flow* apparatus. The former category includes engines, compressors, and pumps having pistons reciprocating in cylinders. These displace definite volumes of the fluid per stroke. The steady-flow types are illustrated by fans, centrifugal pumps, turbines. The fluid flows through them continuously as against the intermittent action in displacement apparatus and the principle of continuity of mass flow, the relative cross-sectional areas of flow, and fluid velocities, generally replace "displacement" in their analysis.

5-2. Slider Crank Chain. This is the basic mechanism of most displacement apparatus. The name is derived from the fact that it is a "chain" of mechanism incorporating a slider and a crank. The chain is pictured as a four-element mechanism in Figure 5-1, the elements being (1) fixed member (frame, base, etc.), (2) crank, (3) connecting rod, (4) slider. A little study will serve to show that this is a mechanism for converting reciprocating into rotary motion and vice versa.

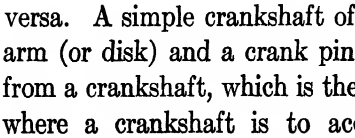


FIG. 5-1. Slider crank chain.

A simple crankshaft of the overhung type is readily made from an arm (or disk) and a crank pin which is set into it at some radial distance from a crankshaft, which is the center of rotation. This type is widely used where a crankshaft is to accommodate but one crank, and where the crankshaft bearing surface is entirely on one side of the crank. Multiple cranks on the same crankshaft are obtained by fitting the crank pins between crank arms, so that the connecting rod may swing freely without interference with the crankshaft. The crank and crankshaft form one of the important basic units of mechanism. There has been found no simpler, more efficient way of transforming rotary to reciprocating motion, or vice versa. It is widely employed in other fields, as well as for heat power engines, and is

paralleled by other mechanisms that can accomplish the same purpose. Nevertheless, through nearly universal usage in displacement equipment, it may be considered *the* basic mechanism.

In displacement equipment the slider becomes a piston and the fixed member is the stationary cylinder and base, or frame. Because it is comparatively easy to form, by usual means of manufacture, and because its shape is very well adapted to the resisting of internal bursting pressure, the cylinder is a very common engineering shape. While, of course, anything of cylindrical shape might truly be called a cylinder, it is customary to apply the term to that part which, in conjunction with a closely fitting internal piston will provide an enclosed space the volume of which may be varied by motion of the piston. Expansion of volume of a working medium is the basis of all commercially employed power cycles, and the cylinder and piston have an important place in this field. Engine-type prime movers have cylinders and pistons. Widespread employment of the internal combustion engine for personal transportation in the United States has made of the cylinder a part familiar to many persons. The cylinder should be made of some hard metal having good wearing characteristics and be machined and honed to a bright finish and exact size. Nickel cast iron, and steel have been extensively used. Cylinders may be made of brass, bronze, or other special materials when the fluid is corrosive to iron or steel. A single cylinder arrangement is shown in Figure 5–2. This illustrates a *single-acting* piston, so-called because only one face of the piston is a working surface. The side next to the crankcase will be exposed constantly to atmospheric pressure. If no natural openings exist into that region, a vent, or "breather," is built into the crankcase wall. This cylinder is shown jacketed to allow either a heating or cooling fluid to be circulated around the cylinder. In lieu of jackets some cylinders will be found with cooling fins, while others are well padded with heat insulation.

The stroke of the piston is always twice the crank radius. Were the connecting rod infinitely long, the percentage stroke accomplished at any crank angle would equal that crank angle expressed as a percentage of 180°. But since rod/crank ratios are usually of the order of from 3 to 8, this simple direct relation is only approximately true. So the reader is reminded that at 90° crank angle the piston is not at mid-stroke, a fact that he may readily verify either by the use of trigonometry or by constructing a scaled diagram. *Displacement* in a piston and cylinder mechanism is the volume swept out by the piston face. It is assumed that the face of the piston is coplanar. Given the bore and stroke as D and L, the number of cylinders n', the piston displacement is:

$$V = \frac{\pi D^2 L n'}{4}.$$

Clearance is the space left in the end of the cylinder when the piston is in dead center position toward the end of the cylinder for which the clearance is determined. This space in volumetric units of measurement is the clearance volume. Clearance is commonly stated as a percentage, being the ratio of the clearance volume to the piston displacement during a stroke. The percentage clearance is a characteristic of no little importance to the performance of displacement apparatus. *Compression ratio* is a term associated with

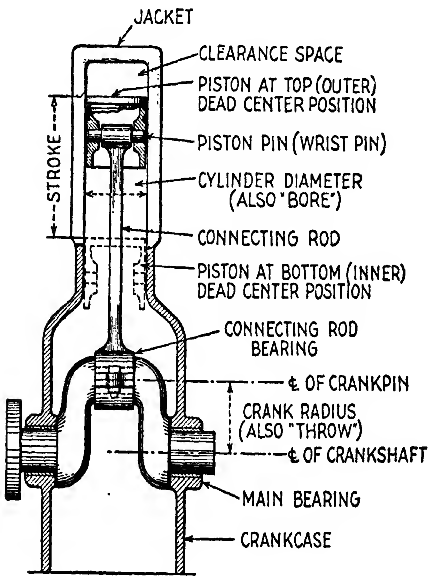


Fig. 5–2. Elements of the single-acting displacement mechanism.

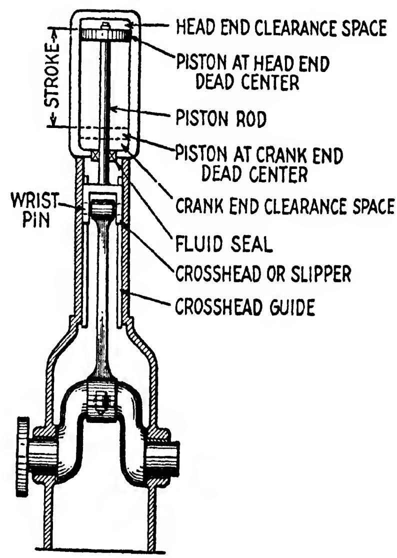


Fig. 5–3. Elements of the double-acting displacement mechanism.

displacement machines. The volume of the cylinder contents when the piston is at bottom dead center (BDC) divided by the volume at top dead center (TDC) is the compression ratio. Calling this term r, and denoting the clearance volume as a percentage C of piston displacement,

$$r = \frac{\text{Piston displacement} + \text{clearance volume}}{\text{Clearance volume}}.$$

$$r = \frac{100 + C}{C}.$$

A *double-acting* machine is shown by Figure 5–3. Both faces of the piston are working faces, consequently both ends of the cylinder must be closed. The swinging connecting rod is not joined directly to the piston because the opening it would require could not be sealed against internal fluid pressure. Instead, there is a piston rod having straight-line motion. One end is rigidly

fixed to the piston, the other to an outside *crosshead*—a sliding member. The connecting rod is swung on the crosshead pin and the hole through which the piston rod enters the cylinder is packed with some kind of fluid seal. Various types of fluid seal are to be described in this chapter. The crosshead takes the side thrust arising from the connecting rod *angularity* so the piston needs very little bearing surface against the cylinder. Note the difference in pistons shown for the single- and double-acting machines. Displacement of the double-acting machine involves the active area of both faces of the piston.

Pistons may be classified as disk, plunger, or trunk type. Most steam engines are double-acting, having the piston joined to the connecting rod by a piston rod and crosshead. The piston is disk-like in shape, being relatively thin, and having the piston rod rigidly attached to it. The steam engine piston is different from the internal combustion engine piston because the latter works at much higher temperatures, and often at higher pressures than the steam engine. The internal combustion engine is usually single-acting, so its piston is of the trunk type. A trunk-type piston is one which has the shape of a short cylinder closed at one end, and open at the other. The closed end forms the piston face, in contact with the working medium. The remainder of it is the skirt of the piston, and is provided to align the piston properly in the cylinder, to support the wrist pin, and to provide bearing area on the cylinder wall against the side thrust arising from the angularity of the connecting rod. Since the gasoline engine operates at a much higher rotative speed than the steam engine, more attention is given to reducing the weight of reciprocating parts, such as the piston, in order to secure a better running balance. Many such pistons are made of aluminum alloy. The typical gasoline engine piston has a length about equal to its diameter. It carries internal bosses for the wrist or piston pin, and is ribbed near the top to give strength to the thin sections used. Leakage past the piston is sealed off with cast iron or alloy piston rings which have the form of slices from a thin-walled cylinder. The rings are split and springy so that in use they press firmly against the cylinder wall, closing the gap between piston and wall. Piston ring grooves are machined circumferentially near the piston face. There are usually two or three of these grooves, the purpose of which is to contain the compression rings. Below them is another groove, which may be of the same size, but sometimes is larger, in which is installed an oil scraper ring, whose function is to prevent excessive amounts of oil being pumped by the piston up to the

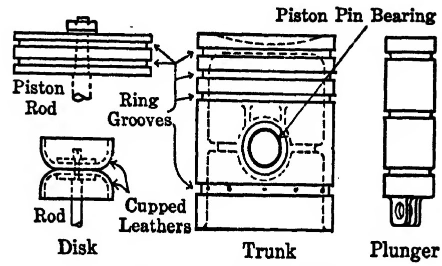


Fig. 5-4. Piston types.

combustion space. The piston pin may oscillate in either the piston bosses or the connecting rod, or may float in both.

Some pistons have specially shaped heads which are designed to carry out some idea related to the shape of the combustion chamber. Some are dished, some are crowned, some are very irregular in shape, but in each case the shape chosen was one which seemed especially suitable in conjunction with the shape adopted for the end of the cylinder.

Double-acting designs are occasionally found in internal combustion engine practice, and such pistons are hollow disks through which cooling water is pumped. The water enters and leaves through a hollow piston rod, to which is attached a swinging or telescoping water inlet tube.

5–3. Steady Flow Apparatus. This category is exemplified by steam and gas turbines, fans, blowers, and rotary pumps. While the theory of physical action of the fluid in these machines is often complex, as a mechanism they are invariably simple. Since flow is steady, there is no need for subdividing the fluid and timing its entry—as must be done in reciprocating machines. Generally there are two principal parts, a rotating member and a stationary one enclosing it and forming an external casing. The rotary member is usually supported on a shaft which, in turn, is carried by two bearings where it protrudes through the casing. The rotary member is variously a wheel, a disk, a drum, an impeller, depending on the machine. These members are so shaped (or carry buckets or blades attached to them), that they create, as by pressure, physical changes in the fluid passing through the device. In them the principle that mass is indestructible is much in evidence, and is frequently employed in the analysis of physical changes of the fluid. In brief, mass, being indestructible, may be completely accounted for at different points of a steady flow, even though physical changes have occurred.

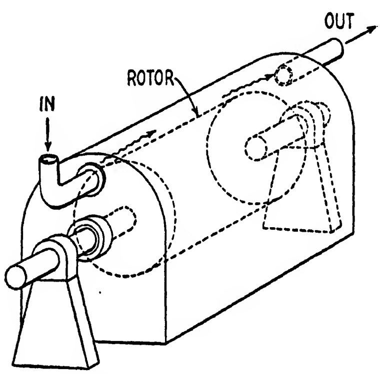


FIG. 5–5. Axial flow machine.

When creation of a centrifugal acceleration is part of the action performed in the fluid, radial flow is used. The fluid is inducted near the center of rotation and flows outward roughly parallel to the radius of the rotating member.* This is typical of centrifugal fans and pumps. Axial flow is found in turbines and some compressors. Here, as is illustrated in Figure 5–5, the fluid enters at one end of the casing and leaves at the other. Action in it occurs in the annular space between casing and rotor. If fluid pressures are high, a seal will be provided at the intersection of shaft and casing.

* Figure 6–7.

5-4. Fluid Seals. Unless a seal is provided the fluid will leak through the clearance, left for mechanical reasons between shaft and casing, if its pressure exceeds that of the surroundings, or it will be diluted by inflow if at a lower pressure. Such seals are not difficult to provide if the shaft has no motion other than axial rotation or reciprocation. Common examples of such seals are (1) the "stuffing boxes" of double-acting reciprocating compressors, pumps, and engines where the piston rod passes through the cylinder, and (2) the stuffing boxes and "glands" of centrifugal pumps and steam turbines where rotating shafts protrude from the casing. Piston rings could be called fluid seals, as could also the method of sealing off fuel gas in large telescoping gas holders ("gas tanks") with a liquid. There are several applications of the floating inverted bell as a gas holder using a liquid seal.

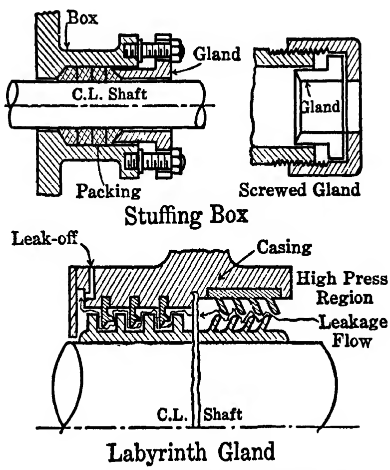


FIG. 5-6. Fluid seals, showing two variations.

A stuffing box is a recess in the wall surrounding the point of exit of the shaft, arranged to receive a soft and pliant packing such as treated hemp, or leather, which is compressed in the box and pressed firmly against the shaft by the pressure exerted against it from a *gland*. The gland is tightened against the packing by screw threads until leakage is reduced to a negligible amount but not enough to produce seizure or excessive frictional heating of the shaft.

Stuffing boxes are used primarily to seal against leakage around small shafts. Leakage glands are preferred for large diameter shafting—or whenever no mechanical contact is wanted in the seal. A typical example is the *labyrinth gland* of the steam turbine. Here the steam is allowed to flow through the clearance space but is given so many turns that the fluid friction and vortices resulting from numerous changes of direction account for the pressure difference across the gland with only a small flow of the fluid through it.

The diagram shows an outflow of leakage steam. When a vacuum exists inside the casing there would be an air inflow through this gland. To prevent this the gland may be designed for admission of high-pressure steam or water to its midpoint, from which there is two-way leakage flow to the edges. Thus water or condensible steam replaces air inflow.

5-5. Cams and Eccentrics. Often the conversion of rotary into reciprocating motion involves special modifications of the straight-line motion. The

slider crank chain can yield only one type of reciprocating motion, known as simple harmonic motion. Also the throw needed is sometimes quite small, and a cam or an eccentric is a more practical way of obtaining it than is the crank and connecting rod.

A cam is a moving part having a surface which transmits a predetermined irregular motion to a second machine part called the follower. While the great majority of cams have a rotating motion, some have an oscillating or reciprocating motion. The cam is employed to give motion to a follower either where the motion is complicated, or where a simple reciprocating motion is better obtained with a cam than with a crank. The cam follower frequently has a reciprocating motion, but not necessarily. The accompanying figure shows several forms of cams of the rotating type, with various types of followers, some of which are reciprocating, some oscillating. The figure also illustrates the simple face cam and the positive motion cam. The followers of the simple face cams must follow the surface of the cam by being held against it either by the force of gravity, or by spring pressure.

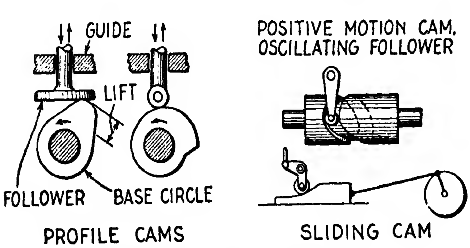


FIG. 5-7. Some cam types.

A cam may be designed to give any desired motion, no matter how complicated it may be. Very often the marvelous features of automatic machinery which may seem to operate with human intelligence are obtained by the use of large numbers of specially designed cams. In addition to complicated motions, simple cams are used to reciprocate followers such as pump plungers, valves, etc., whose motion, while not especially complicated, can be most easily and simply obtained by the use of cams.

The eccentric is likewise an element employed to convert rotating to reciprocating motion. The eccentric is used chiefly for short throws, where it would be undesirable to break the shaft, as is necessary in the case of a crank. It consists of a disk mounted on a shaft in such a way that the geometric center of the disk does not coincide with the center of rotation. The distance between the center of rotation and the geometric center of the eccentric is the "throw." This corresponds to the crank arm distance of an equivalent crank. The eccentric must be used in conjunction with an eccentric strap which surrounds the eccentric, and which

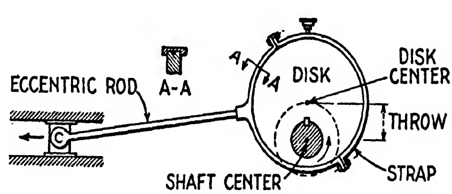


FIG. 5-8. Simple eccentric.

transmits the reciprocating motion to an eccentric rod rigidly attached. The eccentric is chiefly used to drive auxiliaries such as valve gear, and where reciprocation of small magnitude is needed.

5–6. Valves. The cylinders in Figures 5–2 and 5–3 have no port openings shown. Actually the displacement apparatus, whatever it is, must be

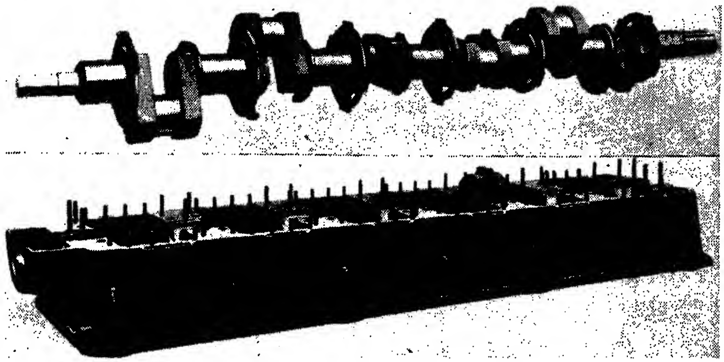


Fig. 5–9. Crankshaft and bedplate of a heavy Diesel engine. (Courtesy Fairbanks Morse & Co.)

provided with port openings. The port is that connecting passage or opening by means of which a fluid flow is admitted to such a region as that furnished by a cylinder. The gasoline engine, the Diesel engine, and the steam engine, as well as the air compressor, have openings arranged in the cylinder so that the fluid may enter and leave properly. In the steam engine, the single port may sometimes serve for both admission and exhaust of the steam. This port leads from the valve chest into the end of the cylinder. The valve will alternately connect the cylinder with either the high-pressure steam or the exhaust region, and the steam flows first in through the port, then out. Some steam engines are double-ported, having separate ports for admission and release of steam. The internal combustion engine usually has two ports, one the inlet, the other the exhaust, since the internal combustion type valve is not suitable for single valve practice. The typical port is a circular opening cored into the cylinder block, terminating at one end in the valve seat, and at the other in the manifold. Usually the valve stem passes through this port for a short distance. If an exhaust port, it must be well water-jacketed or finned, so that heat may be conducted away

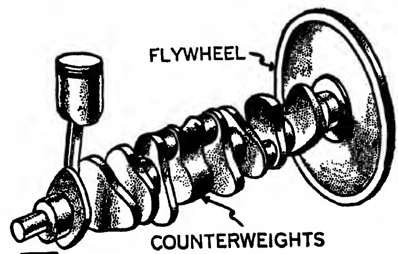


Fig. 5–10. Crankshaft for a high-speed automotive type gasoline engine. (Courtesy General Motors Corp.)

rapidly enough to prevent damage to the metal. Smoothness of stream flow through the ports, and minimum friction loss are desirable in obtaining the maximum possible inflow to the cylinder, or exhaust from it. The size of a port is fixed by the cylinder bore and piston speed.

Engine valves are the closures of the ports of entry or exit. Correct execution of the thermodynamic cycle of operation of a heat engine is dependent upon precision timing of the valve motion, admitting and releasing the working medium.

Valve types may be classified as:

1. *Sliding.* D slide valves, piston valves, and other steam engine types. Sleeve valves.
2. *Oscillating.* Corliss valve, a steam engine type.
3. *Poppet.* Reciprocating, but not sliding. Mushroom and tulip valves of internal combustion engines. Balanced poppet valves for steam engines.

Valves are actuated, and timed relative to the cycle of operation, by the kinematics of the valve gear.

Although engine valves must be timed and actuated by a valve gear, the same does not apply to valves of compressors and pumps. Here the required actuation can be secured from cyclic pressure changes in the cylinder itself, and such valves are simply spring-loaded. When the pressures on opposite sides of a valve are unbalanced sufficiently to overcome the spring pressure it opens until the pressures are either in balance or are in reverse. Poppet valves, ball valves, and flap valves are typical.

There is no more critical part of the I.C. engine than the valves, for these are required to remain pressure-tight under the severe condition of high operating temperature. The exhaust valves of heavily loaded engines operate continuously in an atmosphere of flame, and become red hot. The valves, especially the exhaust valve, which has not the benefit of the cooling derived from an incoming charge, are sources of trouble. By the use of special cooling and of alloy steels, these failures have been greatly reduced in number. The poppet valve is used on nearly all I.C. engines. The only other type to be employed at all is the sleeve valve, in which a sleeve is placed between the piston and the cylinder. The piston slides smoothly in the sleeve, which has some slight motion relative to the cylinder walls. The sleeve is driven by eccentrics or cams, so that a slot in it registers with a fixed port on the cylinder wall when the cylinder is to be opened to the manifolds. Two concentric sleeves are used in the Knight sleeve valve design. Of much more importance from the standpoint of usage, is the poppet valve. This valve has a disk-like head attached to a stem. The stem reciprocates in a valve guide under the action of a cam which bears against the end of the stem, or which operates a

tappet which, in turn, bears against the valve stem. The head has a face which is a portion of a cone, and which sets in a conical seat in the cylinder, or combustion chamber. There is no sliding action between the valve and its seat. Mushroom and tulip type poppet valves are shown in the accompanying diagram. The mushroom type is the simplest, and is suitable for ordinary work. When valves run continuously at very high temperatures, and are made fairly large, the mushroom head has a tendency to break or warp, because its strength is derived by cantilever action, and there is a point of weakness at the attachment of the stem. The tulip valve is in tension instead of bending. It is less liable to leak and gives a better streamlining in the port. This type of valve, when constructed with a hollow stem, partially filled with

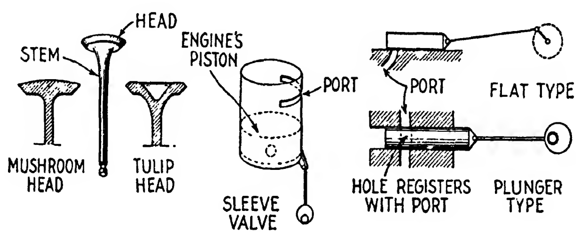


FIG. 5-11. Some valve types.

metallic sodium, which assists in the transfer of heat from the head to the rest of the stem, has been able to perform satisfactorily in the high-output aeronautical type engines. The materials which are used must be alloy steel to withstand the corrosion, abrasion, scaling, burning, and high stress met in service. While carbon steel may be used for inlet valves, chrome nickel or chrome silicon exhaust valves are required. The valve seats are often cut and reamed into the cast iron which comprises the cylinder block of small and medium duty engines. Large valved engines, and engines having aluminum cylinder heads, require inserts of aluminum bronze, stellite, or special alloys, to form the valve seats.

5-7. Valve Gear. The mechanical linkage by which the valves of an engine derive their motion and timing from the crankshaft rotation is called, collectively, the *valve gear*. The term is not restricted to any type of engine but is in more frequent use, perhaps, in the steam engine field. Here the internal-combustion engine gear will be described. Steam engine gear will be found in Chapter 14.

With very few exceptions, the internal combustion engine uses a poppet valve, which is actuated by a short reciprocation, the amplitude of which is well within the ability of a cam to produce. The cam is also an economical method of converting rotation to reciprocation for multi-cylinder engines because of the comparative ease with which a number of cams may be machined on a single shaft. The poppet valve is retained against its seat by a

strong spring. When it is to be opened, a rod or lever is pressed against it, and moved far enough to create the necessary valve lift. This lifting motion

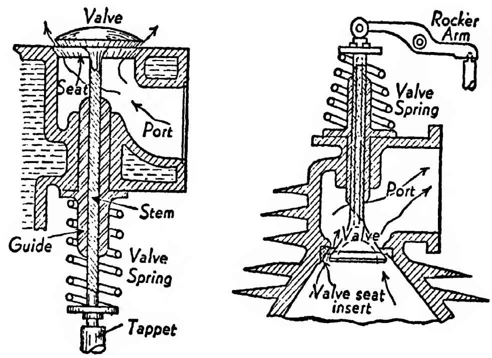


FIG. 5–12. Location and support of poppet valves.

is derived from the cam through a cam follower, and tappet rods or push rods, as illustrated in Figures 5–12 and 5–13. The camshaft itself is driven from the crankshaft. Camshaft drive must be positive, so gears or chains are always used.

Two variations of valve gear suitable for in-line engines are pictured in Figure 5–14. Since these possess nearly the same bore and stroke, the figure demonstrates the effect of valve placement and valve gear space needs upon the engine shape. These sketches are intended to be illustrative of valve gear only and many essential engine details (i.e., valve spring, piston, etc.) are purposely omitted. Figure 5–14A represents a common arrangement of the gear in an L head four-cycle engine. The camshaft is gear-driven from the crankshaft at half speed. The cam bears against a flat follower which raises a tappet, whose pressure against the valve stem then accomplishes the valve opening. The beginning of valve motion is a function of the mesh of the gears and can be altered by placing different teeth in mesh. The height and duration of valve lift are determined by the cam profile. Because of the likelihood of unequal thermal expansion of the valve gear and the cylinder barrel a slight clearance is introduced between tappet and valve stem to prevent the

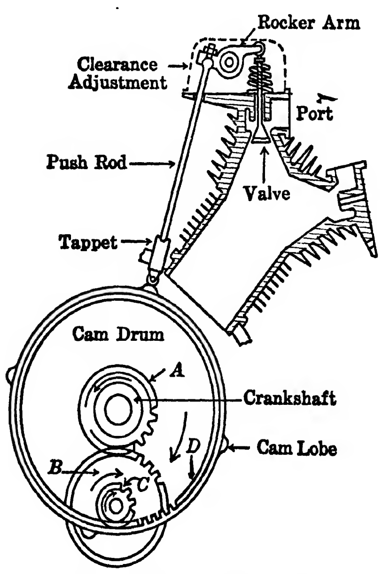
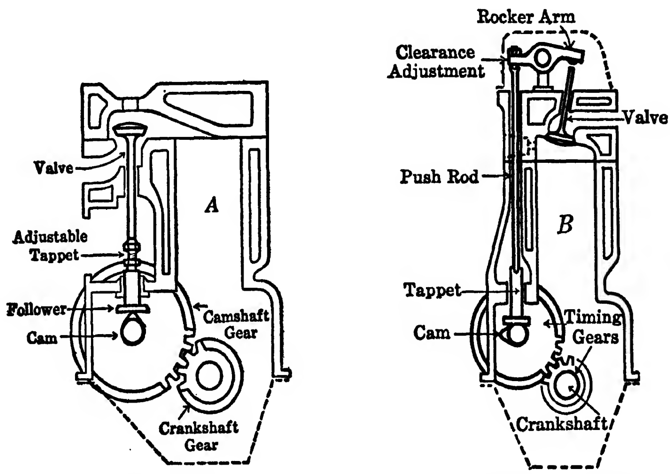


FIG. 5–13. Radial engine valve gear.

valve from riding open continuously when hot. This clearance should be precisely maintained within a thousandth of an inch of the builder's recommendation. Adjustments, either at the tappet or rocker arm (case B) are provided for this purpose. To prevent valve gear noise arising from the cyclical closure of this clearance, special devices such as hydraulic tappets have sometimes been used. These keep the above-mentioned clearance closed when the engine is running but yield and compensate for expansion before the valve will ride from its seat in closed position. The overhead valve (B)



(A) L head engine. (B) Overhead valve engine.

FIG. 5–14. Internal combustion engine valve gear.

requires a few more parts in its valve gear but is often used by engine builders on account of thermodynamic benefits associated with this location.

Some radial engines have been built with a short two-cam camshaft for each cylinder. However, a more common gear for these engines is the multiple lobe cam mounted on a drum (Figure 5–13). A radial engine is built with an odd number of cylinders for each row so that cylinders may be fired alternately (i.e., firing order 1–3–5–7–9–2–4–6–8 for nine cylinders) in the interest of good dynamic balance. A plate cam with four lobes, revolving oppositely to the crankshaft rotation at one-eighth speed will accomplish the operation of the inlet valves necessary for this type of firing order. Another group of four lobes offset from the inlet group will attend to the requisite exhaust valve motion. The gear A is mounted tightly on the crankshaft and turns gears B and C which are integral and mounted on a fixed pivot. C meshes internally with gear D which is cut on the inside of the cam drum. The cam drum has a bearing on the crankshaft. The combined reduction of A, B, C and D is 8:1 and it will be observed that D has rotation opposite to A.

5–8. Governing Prime Movers. Ordinary governing consists of varying the power of a prime mover in accordance with the demands made upon it by the power user. By governor is generally understood the mechanical governor, used to effect a change of throttle position, spark advance, etc. The centrifugal force of rotating masses is the most common principle underlying governors. Fluid pressure produced by fans or centrifugal pumps, the rotors of which revolve with the prime mover shaft, has also been used, but greater dependence is laid upon the principle of rotating masses. A governor of this type is diagrammed in Figure 5–15a. The rotating fork R carries in its ends the pivots upon which are mounted the masses m. By means of the bell-crank linkage and the sliding yoke S, the weight W tends to keep the flyballs

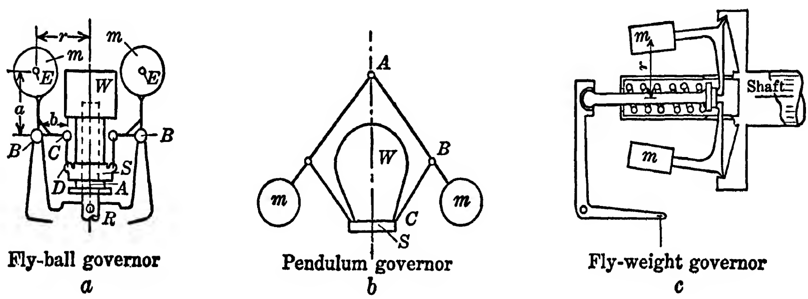


FIG. 5–15. Mechanical governors.

in equilibrium in the position shown under the influence of the mass of the weight, transmitted through the connecting links, and the centrifugal force acting at the center of gravity of the governor weight. Considering that the rotative speed of the governor is ω, the magnitude of this centrifugal force in weight units is:

$$F = mr\omega^2.$$

Heavier, slower speed governors are constructed with the heavier masses arranged as shown in Figure 5–15b, this type being known as the pendulum governor. In general, the higher the rotative speed of the prime mover being governed, the lighter will the governor weights be. Governor weights operated from horizontal shafts act against a spring instead of a dead weight. Figure 5–15c shows governor weights mounted on a high-speed horizontal shaft.

Governor action on internal combustion engines for the purpose of regulating the speed to some normal value is undertaken in a number of ways. For example, the governor may hold the exhaust valve open, preventing any power stroke from being accomplished, or it may interrupt the ignition circuit, causing explosions to be missed by failure to ignite. Or the passage supplying the inflammable fuel air mixture may be throttled mechanically by a valve to

vary the amount of fuel with which the cylinder is charged. In Diesel engines, governing is accomplished by varying the amount of oil pumped into a cylinder during that portion of the power stroke given over to the combustion of a fuel. This is done by a governor which either regulates the stroke of the fuel oil pump, or operates a variable by-pass in the discharge from the fuel oil pump. Governing of a steam engine consists of varying the amount of steam admitted to the cylinder per stroke.

The steam engine may be governed with mass governors of two types, depending on the particular system of governing employed. The smaller, cheaper engines are governed by altering the throttle pressure through imposing an artificial pressure drop created by a governor valve between the boiler and the cylinder. This is known as a throttling governor, and is simplest, but least efficient. It can also be accomplished, still allowing full pressure steam to enter, by varying the portion of the stroke during which the admission valve remains open. This is known as cut-off governing. It is more efficient, but more complicated than the throttled governing. A throttling governor may be simply a flyball type, as already described, arranged so that the motion of the sleeve at *a* can be transmitted through a forked lever to the stem of the governor valve in the steam line. The automatic cut-off type is usually built into the flywheel.

Governing of steam turbines is accomplished by three methods, viz.: (1) throttling at inlet, (2) varying number of inlet nozzles in action, (3) varying duration of full pressure puffs (blasts), of which there are several per second. In addition, some turbines are provided with hand-operated by-pass valves which, by admitting high-pressure steam to low-pressure stages, enable the turbine to carry more overload, though, of course, at reduced economy.

5-9. Lubrication. The importance of lubricants to a civilization using machines to the extent ours does must be very great since almost every machine has one or more parts which are in motion relative to some other part. The points of contact where this relative motion occurs must be protected from friction of the destructive type, that is to say, solid friction, in order, first, to prevent seizure and quick failure of the bearings; secondly, to produce efficient operation through the reduction of friction losses; and thirdly, for smoothness, quietness, and coolness of operation.

The most common form of bearing is exemplified by the support of a horizontal rotating shaft carrying a load of some description. Another class of bearing is that between pistons and cylinders, which, due to the widespread use of the piston-cylinder mechanism, has assumed considerable importance. Furthermore, in this class of lubrication, the conditions of service are the most severe, because of the exposure of the lubricant to the contents of the cylinder, which may be extremely hot, or at high pressure, or corrosive. The instances of friction of plain, flat, sliding surfaces are fewer in number than

the others, but not by any means to be considered rare, as many examples, such as crossheads, could be mentioned.

Lubrication of these various bearings differs as to type and quality of lubricant and method of applying it. The semi-solid lubricants are applied by smearing directly on the bearing surfaces, or indirectly through the medium of grease cups, pressure guns, etc. The liquid lubricants (oils) are supplied to lubricated surfaces mainly by the following means: Oil sprays which require the bearing to be enclosed in a casing, and some means, such as a pump, for filling that space with a spray of oil; pressure streams in which the oil is supplied to the bearing under pressure, through drilled shafts or tubing; dip or splash systems, in which periodic and frequent dipping of some part of the bearing into a pool of lubricant provides intermittent oiling of a satisfactory character; ring oiling, wherein a journal is encircled by a loose ring which dips continuously into a reservoir of lubricant (the slow rotation of the ring under the drag of the rotating journal carries oil from the reservoir up to the bearing by surface adhesion to the ring); wick lubrication utilizing capillary attraction provided by a wick dipping at one end into the oil reservoir, and rubbing against the journal at the other; drip lubrication, which varies from the simplicity of occasional oiling manually from an oil can, to automatic lubrication by sight-feed oil cups, which may be adjusted to drip a certain number of drops of oil per minute to the surfaces which are to be lubricated.

5–10. Multicylinders. Displacement equipment is often designed to have two or more cylinders for the purpose of (1) limiting the bore and stroke dimensions for a required capacity, (2) smoother operation or steadier fluid flow. Many different arrangements of the slider crank chain are possible in multicylinder equipment, particularly in the spark ignition engine.* High output engines with the required displacement subdivided into such as 8, 12, 18 or more cylinders, have smaller reciprocating masses represented by piston and connecting rod, and have overlapping power strokes so that cyclic speed variations are small. Since the main purpose of applying a flywheel to any machine is to steady the speed, there is a reduction of required flywheel size if the total displacement is subdivided among several cylinders. Each cylinder has a piston and a connecting rod, but the number of crankpins may be less than the number of cylinders, due to special arrangements of the cylinders.†

Some common arrangements of cylinders are:

1. In-line. Cylinder axes may be upright or horizontal, the upright arrangement being more favored. If the crankshaft is above instead of below the cylinders, it is termed an inverted in-line arrangement. If

* See Figure 8–7.

† For example, 9 cylinders and but one crankpin in a radial engine.

there are crankshafts *both* above and below, they signify an *opposed piston* design.

2. "V," "X," "W" and other banked arrangements of in-line cylinders. A "V"-8, for example, is two banks of in-line cylinders, four to a bank, all mounted on the same base and delivering power to the same crankshaft. The V arrangement is the most common, the others seldom finding favor outside the aeronautical field.
3. Radial. The cylinders are all in the same plane normal to the crankshaft with their axes radiating out from the crankshaft center like the spokes of a wheel. Double radials would have two parallel planes in which the cylinders are radially mounted. There must be a crankpin for each plane.
4. Angle. One upright and one horizontal cylinder. This is sometimes used on compounded compressors and steam engines. The small high-pressure cylinder is set upright and the bulky low-pressure cylinder is left horizontal.

In addition to having a base, or crankcase, and crankshaft in common, the multicylinder design must have a means of distributing the fluid from a common source to the various cylinders and of collecting it again as it leaves the cylinders. This brings the need of a manifolded conduit. With particular reference to the internal combustion engine in the multicylindered form, the manifold is that part which distributes a common fuel-air mixture uniformly to each of the several cylinders, or which gathers up the exhaust gases after they issue from the cylinders, and combines them into one exhaust stream. Since the great majority of internal combustion engines are multicylindered, and further, since the purpose of multicylinders is to provide more uniform flow of power, the manifolds must be well chosen and correctly patterned, or else they will tend to offset the advantage of the multicylinder arrangement. Suitable manifolding of the high-speed I.C. engine is not easy to secure, yet is most important to the success of that engine.

5–11. Four-Cycle. This is an abbreviated expression for four-stroke cycle. The four-stroke cycle is one upon which any type of internal combustion engine may operate, since it describes, not the thermodynamic aspects of a cycle, but rather the sequence by which the cylinder is charged and exhausted. The four-stroke cycle is described as follows. Beginning with a suction or induction stroke, the cylinder is filled with a fresh charge by the outward * motion of the piston. During the suction stroke, an inlet valve is open. Next, on the return motion, this charge is trapped in the cylinder by closure of all valves leading to and from the cylinder, and is thereby compressed. On the next outward stroke, the power stroke, the fuel is burned or exploded to the

* "Outward" from the valves, but inwards toward the crankshaft.

accompaniment of energy liberation, some of which is made usefully available. The final, or fourth stroke, is a return, or exhaust stroke, during which the contents of the cylinder are exhausted through a port opened by an exhaust valve. Thus the four strokes are suction, compression, power, and exhaust. The advantage of the four-cycle principle is that it gives a full stroke for induction of the fresh charge, and another full stroke for scavenging of the burned gas. In this way it promotes high volumetric efficiency. A disadvantage of the four-cycle principle is the intermittent delivery of power. This contributes to making the four-cycle engine bulky in comparison with the two-cycle,

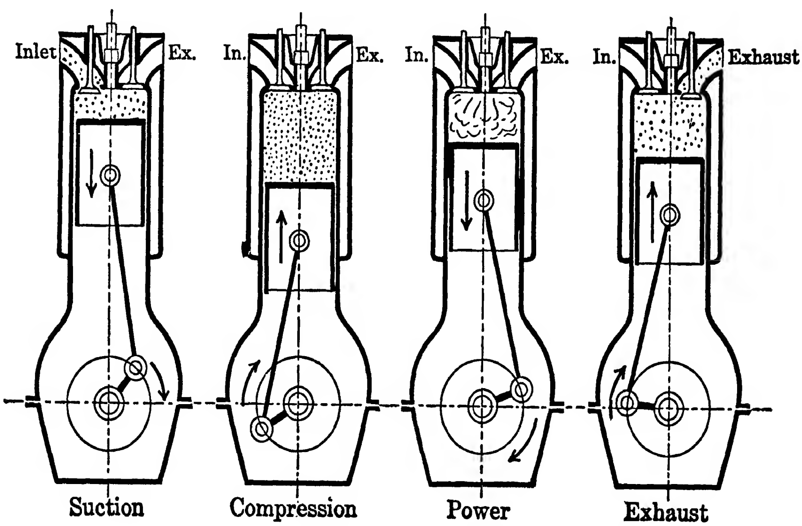


FIG. 5-16. Four-stroke cycle.

but by employing multicylindered engines a steady flow of power may be secured through overlapping of power strokes.

5-12. Two-Cycle. Two-cycle refers to another sequence of operations by means of which the cycle of the internal combustion engine is performed. It is a shortened form of "two-stroke cycle," which may be taken to imply that the cycle is completed in two strokes of the piston.

A two-cycle engine must be so arranged that it can be supplied with a fresh charge when the piston is in the extreme outward position. In general, any internal combustion engine must first introduce the fresh charge, compress it, then ignite and expand it after combustion, obtaining power, then exhaust the products of combustion. To obtain all these functions in two strokes of the piston, it is necessary to shorten the period of time that can be allotted to induction and exhaust. For this reason, although it receives a power impulse every revolution, instead of every two revolutions, the power fails to be double that of the four-cycle engine of corresponding size and speed.

In a simple type of two-cycle engine, valves are replaced by ports in the cylinder, which are uncovered by the piston as it nears crank end dead center. One cylinder port leads to the atmosphere, and through it the burned gases are expelled by the pressure remaining in the cylinder. The other port opens from a by-pass to the crankcase, and through it a slightly compressed charge is delivered to the cylinder when the piston uncovers the port. In some types, the gas is compressed in the crankcase by the piston, but in others an external compressor is employed. The extremely short time which the ports are open renders the two-cycle engine less suitable for high speeds than the four-cycle engine. However, with certain modifications, the speed may be considerably increased without great sacrifice of efficiency. One method is to place auxiliary poppet exhaust valves in the cylinder head, mechanically operated, to aid in clearing the cylinder of burned gas and this will also aid indirectly in obtaining a better induction of the fresh charge. However, the introduction of an exhaust valve nullifies one important advantage of the two-cycle principle, namely, the absence of such valves. Crankcase compression, while simple enough in the single-cylinder gasoline engine, is complicated in multicylinder engines because the crankcase must be separated into sections by air-tight diaphragms so that compression can occur. This offers an obstacle to the smooth supply of fuel to all cylinders from a single carburetor.

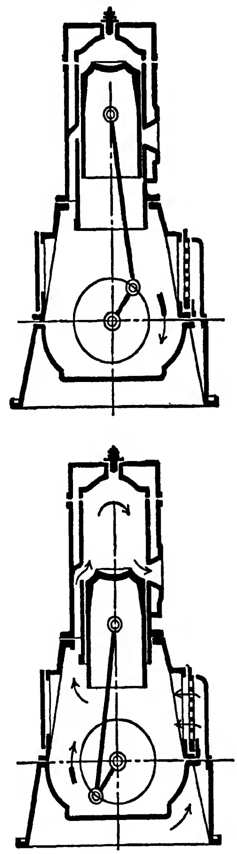


FIG. 5-17. Two-stroke cycle.

5-13. Measurement of Power. The crankshaft is generally the point where power is derived from, or put into, a displacement machine. But it is in the cylinder that the power originates or is expended. Consequently, we may think of *external* power and *internal* power in these machines.

External Power. Shaft power is measured by some form of dynamometer that can be coupled to the machine. Some types of dynamometers absorb all of the power, which is converted into heat, whereas others transmit the power they receive to some other absorber of power, measuring it during the process. These are called, respectively, absorption and transmission dynamometers.

In the *absorption dynamometer* class there are types which convert the mechanical to heat energy through the medium of mechanical friction. They are all similar to the Prony brake. The friction surfaces are variously wooden

blocks against metal drums or pulleys, bands with wooden cleats, ropes, or friction surfaced brake bands. Also there are hydraulic dynamometers which absorb the power by fluid friction. Air friction has also been set to use in the fan brake absorption dynamometers. One of the most convenient means for measuring power is to convert it to electrical form (watts). For an electrical dynamometer, a generator is slightly modified. The stator is mounted on trunnions, but restrained from revolving by a brake arm which is attached to it, and to which are fastened weighing scales. The tendency of the casing to rotate with the rotor which is connected to the source of the power is opposed by the brake arm. The force shown on the scales becomes a torque when multiplied by its lever distance from the center of rotation. Since power is torque multiplied by rotative speed, the only other reading necessary from the dynamometer is the speed of the rotor shaft. In all absorption dynamometers the casing is mounted free to revolve under the action of mechanical friction, fluid friction, or magnetic drag. Actual rotation is prevented by the attached brake arm.

A *transmission type dynamometer* is illustrated by the torsion type, in which a shaft delivering power is twisted through a small angle by the torque. Such a shaft may be calibrated at rest by measuring the torsional deflection obtained under known torque loadings. This dynamometer has its greatest field of usefulness where the other types are impractical. Measurement of power output from a large marine engine is typical.

Figure 5–18 illustrates the common Prony brake, a simple, cheap, easily constructed absorption dynamometer. Engines and motors are often tested with the use of the Prony brake. It consists of the brake drum or wheel mounted on the shaft and surrounded by a band of steel, belting, or some other flexible material, to which is fastened, internally, frictional material. A band-tightening mechanism allows the operator to adjust the pressure between the brake band and the drum. In this way the frictional drag and the load upon the engine can be varied. To the brake band is rigidly attached an arm having a radius r ft. by means of which the band is kept from rotating. The torque effect of the friction drag is found by measuring the force necessary to keep the brake arm from moving. This scale reading in pounds, multiplied by r, is equal to the pound-feet of torque set up as a frictional drag at the surface of the drum. The torque, when multiplied by the angular speed, radians per minute, is the power delivered in foot-pounds per minute. So,

$$\text{Brake horsepower} = \frac{2\pi rFN}{33{,}000}.$$

N is the rotative speed of the drum, revolutions per minute, F is the net weight on the scales, net weight meaning the gross recorded weight less the

force necessary to support the dead weight of the unbalanced brake arm. This deduction is known as the *tare.*

The mechanical energy must be absorbed by the brake itself, as this is an absorption type of dynamometer. For every horsepower 42.4 B.t.u. are generated by friction each minute. The large flywheels of some engines have so much windage or fan action that this heat can usually be dissipated to the atmosphere. Small brake drums suitable for motors must be water-cooled to prevent overheating.

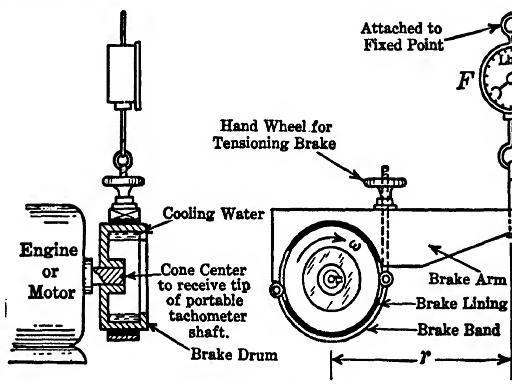


FIG. 5–18. Prony brake dynamometer.

Internal Power. The horsepower developed in the cylinder of any piston and cylinder machine is called indicated horsepower because it can be measured by the indicator mechanism. This power is the result of a gas or vapor pressure pushing against a piston, and is larger than the crankshaft horsepower by the amount of friction and windage losses of the engine.

The engine indicator is an instrument the purpose of which is to give an autographic record of certain of the physical properties of the working medium within the cylinder as the engine or compressor goes through its cycle of operations. A type which traces a pressure-volume diagram is shown in Figure 5–19. The record is traced on a paper of oblong shape which is wound around and fastened to the drum. The lower portion of the drum is a pulley around which is wound a cord. The end of the cord is made fast to the drum, and the other end is attached either directly to the crosshead or through a reducing mechanism. The rotation of the drum under the influence of a pull on the cord is always opposed by a coil spring. In action, the cord transmits to the drum an oscillatory motion which is synchronized with the motion of the piston in the cylinder. This places the paper wound around the drum in a position (with respect to a fixed pencil point) which is proportional to the stroke or to the volume of piston displacement at any instant. The other element of the indicator, the pressure element, consists of a piston moving in a cylinder and opposed by a calibrated spring. The indicator cylinder is connected to the

engine or compressor cylinder by short interconnecting piping through which the instantaneous pressures in the cylinder are transmitted to the indicator piston. The latter rises against the spring pressure until the point of equilibrium is reached. The position of the piston in the indicator cylinder is proportional to the pressure in the engine cylinder. This position is transferred to the record as a corresponding vertical motion of the pencil arm. At the extremity of this arm, the pencil writes on the drum. It draws a record which,

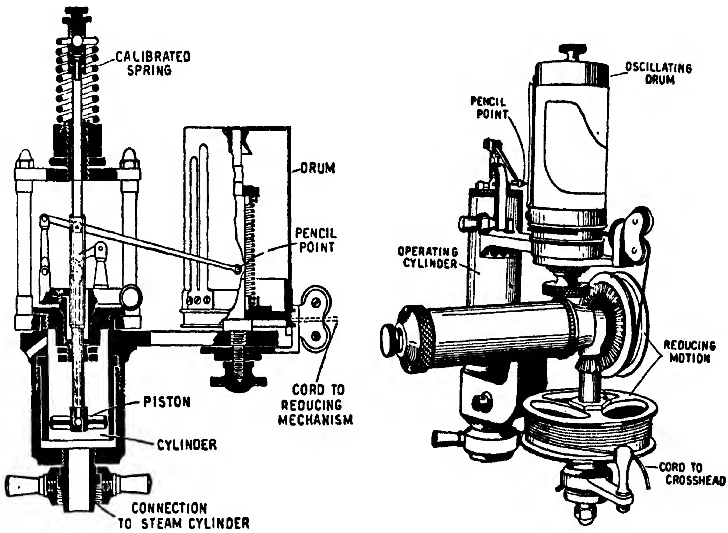


FIG. 5-19. Engine indicators. (*Left*) An "external" spring type. (*Right*) An "internal" spring type with reducing motion. (Courtesy Socony-Vacuum Oil Co.)

by virtue of the features pointed out above, is a diagram of the cycle of pressure vs. the volume in the engine cylinder. Engines with high rotative speed (over 350 rpm.) may not readily be tested with the above described indicator because the inertia of the indicator parts prevents an accurate tracing of the rapidly changing state of the working medium in the engine cylinder. Special high-speed indicators of various designs have been developed which trace a pressure-time diagram. Using data from this curve and the known bore, stroke, and rpm. of the engine, a pressure volume diagram may be constructed.

Effective pressure acting against a piston face and transmitted into a thrust in the piston rod is at any instant the difference between the pressures on the two sides of the piston. In the steam engine, gasoline engine, air compressor, and other piston and cylinder mechanisms, this effective pressure varies throughout the stroke. The average of the effective pressure is known as the mean effective pressure.

If, during a certain power stroke in a cylinder, P is the mean effective pressure acting against each unit of piston area, the piston will receive and pass on to the piston rod a push of $P \times A$ pounds. Here A is the piston area.

When the piston moves through a stroke measured as L feet, the work represented by the motion of the push of PA pounds is PAL. Thus in one minute there are $PLAn$ foot-pounds of work, n being the number of *working* strokes performed per minute. The number of foot-pounds per minute in a horsepower being 33,000, the above explains the origin of the common horsepower formula.

$$\text{Indicated horsepower} = \frac{PLAn}{33{,}000}.$$

The use of this formula requires knowing the bore and stroke of the engine from which A and L are determined, the operating speed in revolutions per minute, and the mean effective pressure P. This mean effective pressure can be obtained experimentally with the use of the indicator. The value for n is determined by consideration of number of cylinders, rpm., whether single- or double-acting, and whether two- or four-cycle. The average height of the indicator diagram is the mean effective pressure. If a machine had no mechanical or fluid friction, and no work were necessary to operate its auxiliaries (such as governor, lubricating oil pump, etc.,), the internal and external power should be the same. Since these do exist, the internal and external power are different. If the external power is an *input* the internal power will be smaller, and vice versa for external power as an *output*. The ratio of these two powers is the *mechanical efficiency* of the apparatus, while their difference is the *frictional horsepower*.

Example 1: A six-cylinder, four-cycle, $3\frac{1}{2}$ in. $\times$ $4\frac{1}{4}$ in. automotive engine revolves at 2800 rpm., with cylinder mean effective pressure of 110 psi. What is the internal power developed?

This class of engine is invariably single-acting. The number of power strokes per minute is determined by starting with the knowledge that in four-cycle action each cylinder receives a power stroke every fourth stroke—that is to say, every other revolution. So each cylinder has 1400 power strokes per min., or 8400 per min. for the engine. The pressure can be used in square inch units if the piston area also is, but stroke must be in feet. We set up the internal, or indicated, horsepower equation:

$$IHP = \frac{110 \times \frac{4.25}{12} \times \frac{3.5^2\pi}{4} \times 8400}{33{,}000} = 95.5 \text{ hp.}$$

Example 2: A Prony brake on a steam engine loaded the engine, producing a speed of 250 rpm. when 155 lbs. were registered on the scales. Preliminary measurements established the brake arm distance as 52 in., the tare as 15 lbs. What output power was being developed?

Substituting in the brake horsepower equation,

$$BHP = \frac{2\pi(155 - 15) \times \frac{52}{12} \times 250}{33{,}000} = 29 \text{ hp.}$$

Notice that neither cycles of operations, numbers of cylinders, nor even type of prime mover enter into the determination of brake horsepower.

5–14. Rating. The output of mechanical equipment is rated on different bases, the particular method being associated with the conditions of usage, or with the historical development of the apparatus.

A few examples of the determination of rating in the mechanical field will show how varied the picture is, and demonstrate that technical equipment is not subject to uniform systems of rating. Steam boilers are rated by a unit based on their heating surface, but progress in the art of generating steam in boilers, and in the field of combustion engineering, has so increased the steam production possible from a given area of heating surface that the actual performance of steam boilers can nowadays be related to their ratings only by the use of a term "per cent rating." I.C. engines could be rated at the peak of their speed-power characteristic. However, some of them (as in aviation), cannot stand up continuously under the maximum output, and so are "rated" more conservatively.

The power rating of a steam engine depends upon the extent to which steam is used expansively in the engine. The less the expansion, the greater the power rating of a given cylinder volume, and the less economical the engine is of steam. Common practice rates the power as that derived when the ratio of expansion is 4. A steam turbine has its nozzles, blades, and passages designed for a certain full load at which steam consumption will be minimum. Loads above this can be carried by admitting some high-pressure steam into low-pressure sections of the casing by overload valves. Naturally the generator to which the turbine is connected must be provided with sufficient capacity to absorb this power.

Air compressors are usually rated by the cubic feet of air per minute they can compress, the air volume being taken at standard atmospheric conditions. Refrigerating compressors are rated in "tons" of refrigerating capacity of the plant they will serve. And so it goes; in each field of applied energy the customary rating unit reflects the special usages.

PROBLEMS

1. Draw, to scale of 1 in. = 10 in., skeleton diagrams of a slider crank chain of 30 in. stroke, having a rod to crank ratio of 3. (a) Slider at 20% stroke, (b) slider at 50% stroke.

2. In a certain slider crank chain the crankpin travels in a circle of 4 in. diameter. The connecting rod is 6 in. long. By graphical method (full scale) determine the degrees of crankpin displacement from TDC for 50% stroke and for 80% stroke of the slider.

3. A two-cylinder, reciprocating, air compressor's leading dimensions are 6 in. × 8 in. × 150 rpm. (bore × stroke × speed). What is its displacement, cubic feet per minute? What would the displacement have been in case the compressor were double-acting?

4. A certain class of racing engines is to be limited to 200 cu. in. displacement. If it were intended to build an in-line engine approaching as near to the upper limit as possible, being 3-in. bore and $4\frac{1}{2}$-in. stroke, how many cylinders should there be?

5. In gasoline engines an increase of compression ratio increases efficiency but invites trouble from detonation (knocking). Suppose the limit is a compression ratio of 6. What is the per cent clearance? Diagram a single-acting 4 in. $\times$ 5 in. (bore $\times$ stroke) cylinder having this compression ratio. Rod to crank ratio $3\frac{1}{2}$. Construct a figure similar to 5–2 but at 50% stroke, to a scale of 1 in. = 2 in. Make the piston "square" with piston pin at mid-height.

6. The clearance of a single-acting ammonia compressor is 30%. What is the ratio of compression? To scale of 1 in. = 2 in., construct a diagram similar to Figure 5–2, but with piston at 40% stroke, showing this compressor. Bore 4 in., stroke 4 in., rod/crank 4.

7. A 10 in. $\times$ 12 in. $\times$ 250 rpm. double-acting steam engine has 5% clearance each end. Rod/crank 6, piston $2\frac{1}{2}$ in. thick, piston rod 18 in. from ℄ piston to ℄ wrist pin. Construct to scale of 1 in. = 10 in., a diagram similar to Figure 5–3, showing the engine at 90° crank angle. (Neglect effect of piston rod on crank end clearance.)

8. In a certain 6 in. $\times$ 8 in. $\times$ 700 rpm. double-acting air compressor the clearances are equal, and the cylinder is 12 in. long, inside. Piston 2 in. thick, rod/crank 3, piston rod $1\frac{1}{2}$ in. diameter, distance from ℄ of piston to ℄ wrist pin 13 in. Construct to the scale of 1 in. = 4 in. a diagram similar to Figure 5–3 for HEDC, except draw in a stuffing box form of piston rod seal. Calculate effect of piston rod on crank end clearance. (Let x = head end clearance distance and y = crank end clearance distance. Two equations involving x and y can then be written.)

9. Draw, full scale, an eccentric which will give a slider block a reciprocation of 1 in. Shaft diameter $1\frac{1}{2}$ in.

10. A disk 6 in. in diameter is mounted on a 3-in. shaft so as to leave a minimum thickness of $\frac{1}{2}$ in. around the hole in the disk. Make a full scale drawing showing this used as an eccentric. If it were employed to operate a steam engine valve, how much travel would the valve have, back and forth?

11. A mushroom type poppet valve with 2-in. (mean seat) diameter and 45° seat angle is lifted $\frac{1}{2}$ in. by a simple symmetrical face cam constructed on a 1-in. diameter base circle. Sketch the valve head and the cam, full scale.

12. Draw the cylinder head (only) of Figure 5–14B full scale, assuming all necessary dimensions except the following: Bore 4 in., valve diameter (face) $1\frac{1}{2}$ in., seat 45°.

13. If gear A, Figure 5–13, had 20 teeth and gear D 200 teeth, how many teeth would gears B and C have, assuming that radii to pitch circles of A and D are 2 in. and 8 in., respectively?

14. If the plate cam of Figure 5–13 had 3 lobes instead of 4, what would be its required speed relative to the crankshaft? Repeat problem 13 for this case.

15. Diagram the valve gear (only) for an overhead valve, four-cycle I.C. engine. Label fully.

16. Diagram and label valve gear of a V-type, L-head, four-cycle engine.

17. Diagram a four-cycle engine in (a) the power stroke, (b) the exhaust stroke. Skeleton diagrams, including overhead valves, but excluding valve gear. Label fully.

18. Diagram a four-cycle engine on (a) the suction stroke, (b) the compression stroke. Skeleton diagrams, including valves in L-head cylinder, but omitting valve gear. Label fully.

19. Diagram and label a two-cycle engine with piston located for the instant of beginning of exhaust.

20. Diagram and label a two-cycle engine with piston located for the instant of beginning of compression. Also, show location (dotted) of piston at the end of induction. No supercharge.

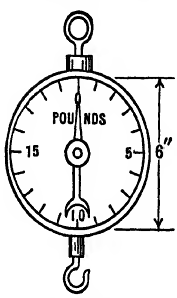
21. A displacement machine on test has a m.e.p. of 52 psi. Bore and stroke are 3 in. × 4 in. It is single-acting, two-cycle, two-cylinder, operating at 300 rpm. (a) What is the internal power, (b) given a mechanical efficiency of 90%, what would the external power be if this were an engine? A compressor?

22. Calculate the indicated horsepower of a single-cylinder, double-acting, two-cycle machine 10 in. × 15 in. × 200 rpm. Mean effective pressure 80 psi. Allow for 2-in. piston rod by computing crank end separately from head end and using *net* piston area.

23. What is the indicated horsepower of an eight-cylinder, four-cycle, single-acting I.C. engine? Dimensions 3 in. × $4\frac{1}{4}$ in. × 2500 rpm. Mean effective pressure 95 psi.

24. A two-cylinder, double-acting air compressor has indicator cards 3 in. long, with average area of 6.75 sq. in. Scale of spring in the indicator 60 psi. per in. on the card. What was the indicated horsepower? Compressor dimensions 6 in. × 10 in. × 150 rpm. Neglect piston rod area.

25. An engine type of displacement machine developing an internal horsepower of 22 at 500 rpm. overcomes an external torque of 2600 pound inches. Find the mechanical efficiency.



26. A Prony brake test gave data as follows: Length of arm 36 in., scale reading 58 lbs. Tare 7.5 lbs., speed 1200 rpm. Find the brake horsepower.

27. A Prony brake with 1.5 lbs. tare weight and 24-in. arm is to be used with a dial type spring balance to determine the power output of a constant speed, 1800 rpm., motor. Supply a chart (full scale) to replace the weight scale—one that will read horsepower directly. Assume that the pointer is adjustable so that zero weight and zero horsepower will coincide.

28. Repeat problem 27 with alternate data as follows: 1200 rpm., 18-in. balanced arm (no tare).

29. A four-cylinder engine with cylinder dimensions of $4\frac{1}{2}$ in. × 5 in. when tested at 1500 rpm. yielded a dynamometer reading of 85 hp. What is the mechanical efficiency for a computed indicated horsepower of 92.5?

30. How many power strokes per minute are there in each of the following machines?

a. Five-cylinder, four-cycle aircraft engine, 2000 rpm.
b. Two-cylinder, double-acting, steam locomotive. 8-ft. drivers, 60 mph.
c. Three-cylinder, two-cycle, single-acting, marine engine, 450 rpm.
d. Eight-cylinder, two-cycle, double-acting, Diesel engine, 350 rpm.

CHAPTER 6

Air and Energy

6–1. Air Compression. Air, being the most abundant form of a natural gaseous substance, and being, on the whole, without serious faults as a compressible medium, is frequently employed in the compressed state. Industrial and commercial uses of compressed air are exceedingly numerous. The compression of air by mechanical means, and the raising of it to some desired pressure above that of the atmosphere, is effected, usually, by an approximate adiabatic change of state.

Ideal adiabatic compression would be represented by the following equation, relating pressure and volume:

$$PV^{1.4} = C.$$

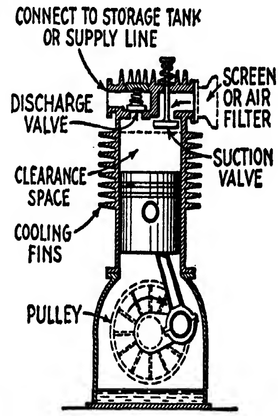


Fig. 6–1. Diagram of a single-acting air-cooled compressor.

A compression of this nature could heat the air to temperatures which would interfere with reliable action of an air compressor and introduce lubrication difficulties, were there no provision for cooling the cylinder walls. Therefore, in compressors we find the cylinders to be externally finned or water-jacketed so that sufficient cooling is secured to keep the temperatures from becoming excessive. The extraction of heat from the cycle in this way modifies the conditions of compression from the ideal to some change more nearly represented by

$$PV^n = C,$$

in which n usually lies between 1.35 and 1.4. It also reduces the work required for compression. The ratio of the temperature before and after compression is expressed by the following equations,* the temperatures being in degrees Rankine.

$$\frac{T_2}{T_1} = \left[\frac{V_1}{V_2}\right]^{n-1} \quad \text{and} \quad \frac{T_2}{T_1} = \left[\frac{P_2}{P_1}\right]^{(n-1)/n}.$$

If the heat of compression is removed by cooling as rapidly as it is formed, an isothermal compression will be had. Less work will be needed for the compres-

* These can be derived from algebraic manipulation of $PV = wRT$, and $PV^n = C$.

sion of a pound of air to the same discharge pressure. Although isothermal compression is desirable it is not possible to achieve it in fast-moving compressors. Even in finned or jacketed cylinders the compression is more nearly adiabatic than isothermal. Actual performance of compressors is, for some purposes, referred to isothermal compression and, for others, to adiabatic compression as a standard.

In compression to high pressures, the temperature rise may be too great to permit the compression to be carried to completion in one cylinder, even

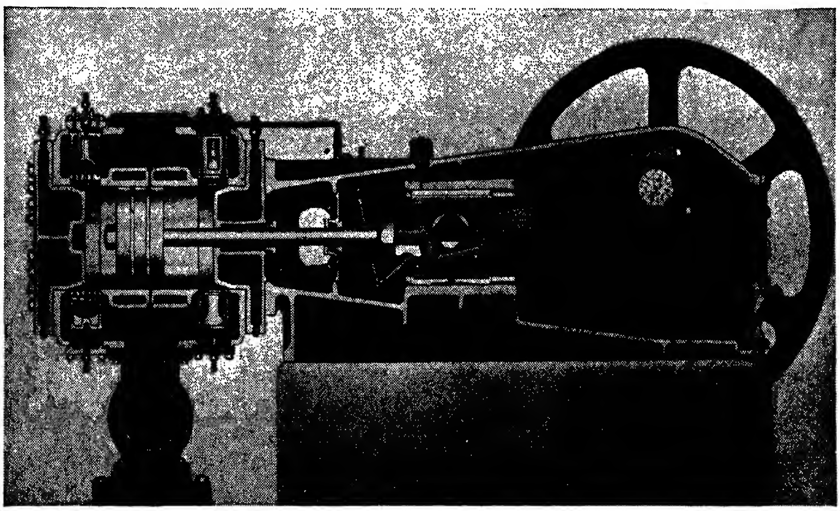


FIG. 6–2. Water-cooled double-acting air compressor. (Courtesy Gardner-Denver Co.)

though it is cooled. In such cases, the compression is carried out in *stages*, with a partial increase of the pressure in each stage, and cooling of the air between the stages. Two- and three-stage compression is very common where pressures of 300–1000 lbs. per sq. in. are needed.

The mechanical construction of air compressors varies with the amount of compression required. Piston and cylinder compressors are usually employed for the highest pressures. In small sizes these are frequently single-acting, but are made double-acting in larger sizes. Moderate pressures may be produced by multi-staged centrifugal action. Light pressures, such as are required in draft and ventilating systems, are obtained most economically by the use of fans of the centrifugal or propeller type.

There is a type of compressor intermediate between the fan and the piston types. It is the rotary compressor or *blower*, which, operating by displacement, produces a positive air pressure, and, at the same time, is a compressor in which there are no reciprocating parts.

The compression of air and other gases may also be secured by the employment of steam jets if an admixture of vapor in the compressed gas is not objec-

tionable. High-pressure steam is blown through nozzles which create a high velocity jet. The gas to be compressed is led into the regions about the nozzle discharge where it is entrained in the steam jet. The mixture then travels into a diffuser for velocity reduction and attendant compression. Although the compression thus achieved is of limited magnitude, staging the compression in a series of nozzles, with intermediate coolers for partial condensation of vapor, allows larger compression ratios to be achieved.

6–2. Reciprocating Air Compressors. A common type of air compressor is the piston and cylinder compressor in which a reciprocating piston positively displaces the air from a cylinder during its discharge stroke. Compressors

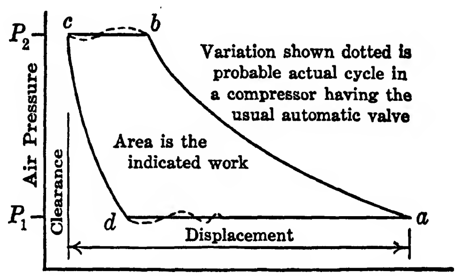


FIG. 6–3. Compression cycle of the reciprocating compressor.

for charging tanks of air used to inflate pneumatic tires at the numerous automotive service stations are of the reciprocating type. Being of small capacity they are generally single-acting and air-cooled (by exterior fins) since those features are common in small compressors. Larger compressors are usually double acting and frequently cooled by water jackets. One- or two-cylinder arrangements are conventional, with a tendency to secure large capacity by increased bore and stroke rather than by multiple cylinders. Pistons are reciprocated by a crankshaft and connecting rod mechanism commonly receiving motion from the driving source by belt. Valves are spring-loaded to open upon slight differential pressures.

The compressor cycle is shown in Figure 6–3. The discharge stroke which begins at *a* builds up the pressure to *b* where it exceeds the receiver pressure sufficiently to open a discharge valve. Discharge then takes place at constant pressure from *b* to *c*. The volume *c* is the clearance volume of the compressor. The air in the clearance volume must expand to *d* during the suction stroke before the inlet valve will open. Thus the volume of air drawn in per stroke is only that from *d* to *a*. Obviously the compressor should have as small a clearance as possible in order to obtain good volumetric efficiency, especially at high discharge pressure. Small compressors may be operated with high compression ratios (8–12) if desired because cooling is more effective in small cylinders and mechanical strength is readily provided. Large volumes com-

pressed to ratios exceeding 4 will need a multi-stage compressor to permit cooling between stages and to lessen the structural loads on the large first stage cylinders.

Mechanically, compressors consist of the following elements:

1. Air cylinders, heads and pistons, together with air inlet and discharge valves make up the compressing element.
2. A system of connecting rods, piston rods, crossheads, crankshaft, and flywheel for transmitting the power developed by the driving unit to the air cylinder pistons.
3. A self-contained lubricating system for bearings, gears, and cylinder walls, including a reservoir or sump for the lubricating oil, a pump or other means of delivering oil to the various parts, suitable filters and coolers. On many compressors a separate force-feed lubricator is installed to supply oil to the compressor cylinders.
4. A cooling system for removing heat from the cylinders and heads, intercoolers and aftercoolers, and lubricating oil.
5. A regulation or control system designed to maintain the pressure in the discharge line and receiver within a predetermined range of pressure.
6. An unloading system which operates in conjunction with the regulators to reduce or eliminate the power used in the air cylinders.

Air valves are the vital part of a compressor. Inlet and discharge valves of present-day compressors are of the automatic type, i.e., the opening and closing of the valves is caused solely by the difference in pressure between the air within the compressor cylinder and the external air on the opposite sides of the valves. Thin plate, low lift valves are the preferred type and are now used in all compressors except for some high-pressure cylinders. They have large areas for the passage of air which permit a low velocity of air through the valve, thus improving the compression efficiency. They open with minimum resistance and close promptly, thereby keeping the range of pressure inside the cylinder within limits close to that for which the compressor is designed; furthermore, they are quiet in operation and simple to replace. These valves are made in a variety of forms such as annular disks, thin strips, and flat plates, and are known by several names such as "plate," "feather," "wafer," etc. Modified poppet valves of the automatic type are still employed in high-pressure cylinders.

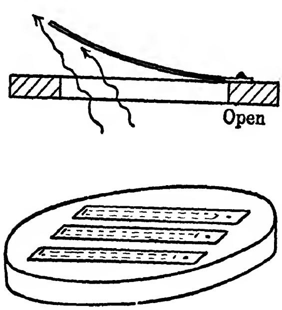


Fig. 6-4. Principle of the thin plate valve.

6-3. Rotating Air Compressors. Rotating compressors may be subdivided into three classes, known as blowers, centrifugal compressors, and turbocompressors.

Blower. Blowers are defined as machines for compressing air at pressures up to 35 lbs. per sq. in. gage. Centrifugal and turbine type compressors may be built to deliver high pressures by providing sufficient stages; however, reciprocating compressors are commonly employed for pressures exceeding 50 lbs. per sq. in.

Blowers are generally positive displacement in action. One type is called the rotating vane compressor. As diagrammed in Figure 6–5, it consists of a cylindrical casing with a cylindrical rotor off center in the casing. The rotor

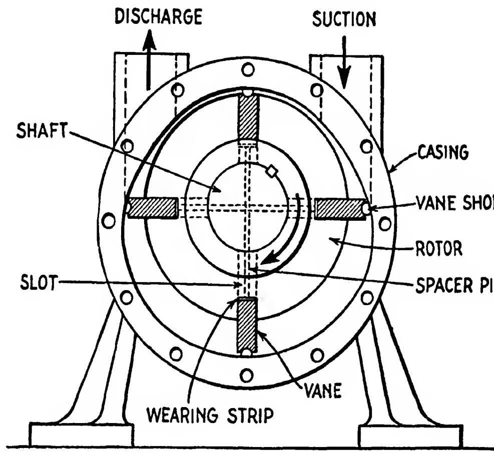


FIG. 6–5. Rotary vane compressor. (Front cylinder head removed). The center of the rotor is offset in a cylinder having a cam-shaped internal bore. This shape is such that all lines passing through the shaft center from inside to inside surface of the cylinder are of the same length. Since the distance between sliding surfaces of any pair of opposite vanes is always the same, rigid spacing pins can be inserted between vanes whose contact with the cylinder walls does not then require springs or depend on centrifugal force. (Courtesy Foster Pump Works.)

carries a number of vanes in radial slots, these bearing against the inside of the casing under the action of springs, or of their own centrifugal force. Although it would seem that wear on the tips of the vanes would be excessive, such is not the case if good internal lubrication is provided. As the vanes pass the inlet opening the volume that adjacent vanes cut off inside the casing is on the increase and air is drawn into the region between vanes. Rotation carries the air around to the outlet where it is squeezed out because of the approach of the rotor body to the casing. Another class of blowers might be termed the "lobe-type blower." This is shown in Figure 6–6. The blower has a cylindrical casing into which are fitted two rotating lobes or rotors. Each has three lobes which fit together like gear teeth. As the two rotors are driven around, air is drawn into the casing and carried around the outer side of the rotors to the outlet. The rotors have been built with straight lobes, but modern designs are twisted into spiral form to give a uniform discharge. The rotors must not rub on the casing but the clearance must be extremely

small or else excessive "slip" will be encountered. Efficiency would suffer and pressure would not be increased sufficiently. To prevent wear, meshing gears are mounted on the rotor shafts. Thus one rotor, when driven from an external source, drives the other but the lobes are not in actual contact. Like the rotor-casing clearance, the lobe to lobe clearance must be extremely small. All this requires a high grade of machine work by the manufacturer.

Centrifugal Compressor. A rotating impeller mounted in a casing and revolved at high speed will cause a fluid which is continuously admitted near

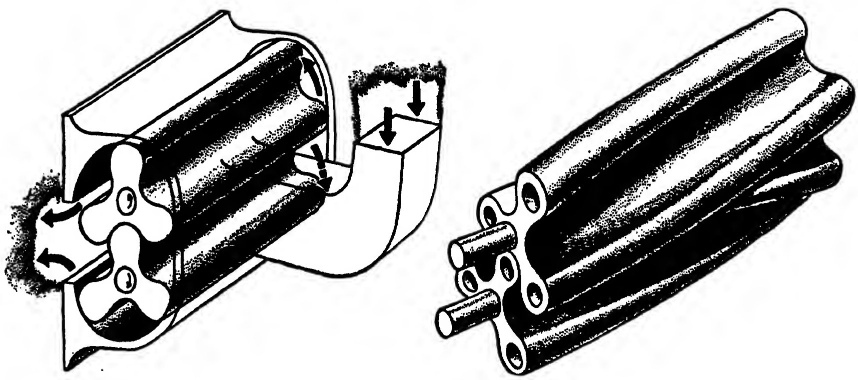


FIG. 6–6. Positive displacement blower-lobe type. (Courtesy General Motors Corp.)

the center of rotation to experience an outward flow and a pressure rise due to centrifugal action.

Assume that an impeller with radial blades of depth $r_2 - r_1$ is revolving at a speed of ω radians per minute. This is illustrated in Figure 6–7. Consider that a compressible fluid (a gas) is admitted at the center and flows into the impeller radially. Relative to the impeller blades it has an outward radial flow, finally emerging with some absolute velocity v_2 which is partially diffused into pressure. In addition a pressure gradient must exist to balance the sum of all the incremental $mr\omega^2$ inertia forces arising from the inward acceleration $r\omega^2$ given each particle of the fluid. The interrelation of r, ω, P, can readily be developed by considering the power required as (1) that necessary for the thermodynamics of compression, and (2) that which would account for the action of the impeller in effecting certain momentum changes on the fluid.

(1) Assume a flow of one pound per minute. Then, from page 138, the power required for an ideal adiabatic compression and delivery is RT_1/z $[(P_2/P_1)^z - 1]$ ft. lbs. per min. Although there may be no transferred heat,

it is likely that the power for an actual compression will exceed the above on account of turbulence and friction in the gas being whipped around by the impeller. Let η_c represent the internal efficiency.

(2) Now view the impeller action as mechanical. The power input to the impeller is used to alter the momentum of the gas flowing through it. As v_1 was assumed radial, it has no initial tangential momentum. Next examine

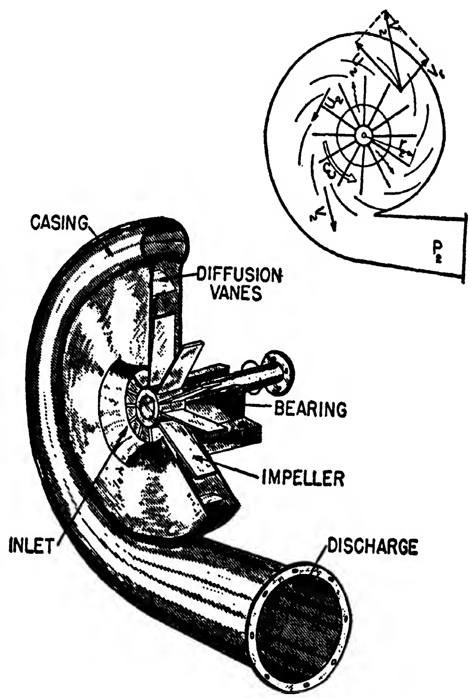


FIG. 6-7. Centrifugal compressor.

the momentum at r_2. The velocity v_2 is the resultant of a radial relative velocity v_r and a tangential rim velocity u_2. Call the tangential, or whirl, component w_2. The tangential momentum at r_2 now appears to be mw_2. This demands an input torque equal to mw_2r_2. Again assuming a flow of one pound per minute.

$$\text{Power} = \omega \times \text{torque} = \frac{w_2 r_2 \omega}{g}.$$

However, as

$$\omega = \frac{u_2}{r_2}; \quad \text{Power} = \frac{u_2 w_2}{g}.$$

The vector diagram in Figure 6-7 shows that $w_2 = u_2$ if v_r is radial, hence

$$\text{Power} = \frac{u_2^2}{g}.$$

Next equate these two expressions for power:

$$\frac{RT_1}{\eta_c z}\left[\left(\frac{P_2}{P_1}\right)^z - 1\right] = \frac{u_2^2}{g}$$

or

$$\frac{P_2}{P_1}* = \left[1 + \frac{\eta_c z u_2^2}{gRT_1}\right]^{1/z}.$$

Typically, η_c will range from .75 to .85.

If the inflow is not radial, and if the blades are curved, the action is not so readily analyzed. For pressure ratios higher than can be obtained by the

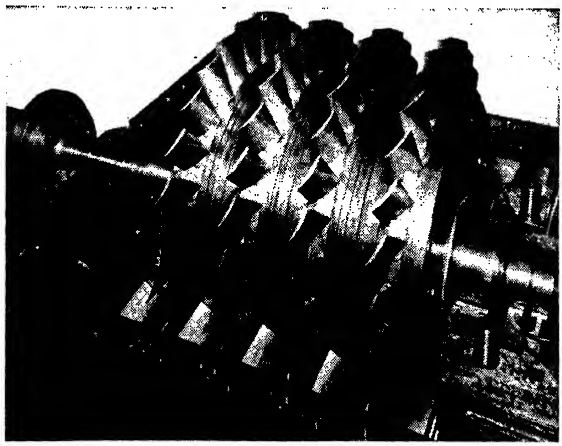


FIG. 6–8. Axial flow compressor (without stator top half). (Courtesy General Electric Co.)

action defined in this equation, several impellers may be mounted on the same shaft and enclosed in a compound casing with passages arranged to lead the output from one impeller to the "eye" of the next. This multi-staging principle is used to produce pressures above the capability of a single impeller compressor. Multi-stage compression may have cooling between the stages so the overall compression may be more isothermal than adiabatic.

Turbo-compressor. This is a multi-stage axial-flow compressor, so called because of the resemblance, in action, to a reversed turbine. Figure 6–8 shows a turbo-compressor. Air flows over a set of airfoils (arranged circum-

* It is interesting to find that if centrifugal action, only, is considered (as for discharge closed), the result is:

$$\frac{P_2}{P_1} = \left[1 + \frac{\eta_c z u_2^2}{2gRT_1}\right]^{1/z}.$$

This implies that centrifugal force accounts for half the energy input, diffusion the other half.

ferentially as blading). The airfoil blades turn the air stream through an angle. A diffusion thus effected slows down the air velocity and increases pressure. The blade heights are the same in all stages of the compressor, Figure 6–8, because slower air speeds were used in succeeding stages; however, sometimes air speeds remain nearly constant, then blade heights can diminish in succeeding stages (as witnessed in Figure 10–8) owing to increasing density of the compressed air. Operating the blades at high angles of attack on the air helps build up pressure rapidly, but operation near the stalling angle would be hazardous as small variations might occur which could burble the airfoils and cause an unstable, rough, or even hazardous condition to exist. Secondly, turbo-compressors may be employed under conditions where utmost efficiency is imperative (as in gas turbine power units) and should create the optimum favorable balance between good downwash and minimum turbulent airfoil wake. Turbo-compressors have been built with energy efficiencies as high as 85%. They may be operated effectively at high speeds, i.e., 5000–10,000 rpm.

6–4. Work of Air Compression. In order to produce compressed air the piston of a compressor executes a working stroke part of which (*ab*, Figure 6–3) is a non-flow polytropic *compression* of the air occupying the cylinder at state *a*, the remainder of the stroke being *delivery* at constant pressure (*bc*). On the return stroke the clearance air does work by re-expanding from *c* to *d*. For purposes of comparison a cycle with adiabatic compression and zero clearance is here * made a reference standard, and the performance of actual compressors referred to it with a *compression efficiency*,

$$\eta_c = \frac{\text{Ideal work of compression}}{\text{Work actually required for compression}}.$$

η_c is also called "adiabatic efficiency," and "internal efficiency." As originally defined, η_c is based on cylinder, or *indicated* work, so that were the compressor mechanical efficiency η_m, the input work would be the ideal compression work divided by $(\eta_c \times \eta_m)$. The product $\eta_c\eta_m$ should be called the *compressor efficiency*, but η_c is also often loosely used to denote overall compressor efficiency.

Since cooling fins and jackets reduce the volume of each pound of air delivered from the cylinder, compared to adiabatic compression, they are an aid to securing high compression efficiencies. If the gas were so thoroughly cooled that compression were nearly isothermal, it could be possible to achieve compression efficiencies of more than 100%. The work required for *adiabatic* †

* This practice is not universal. Some prefer isothermal compression as a standard of comparison.

† The corresponding expression for *isothermal* compression is $W = RT_1 \log_e P_2/P_1$.

compression and discharge, per pound of air compressed is:

$$W = \frac{RT_1}{z}\left[\left(\frac{P_2}{P_1}\right)^z - 1\right] \text{ ft. lbs. per lb. air,}$$

in which W = Work done on an ideal compression.

$z = (\gamma - 1)/\gamma$.
T_1 = Suction temperature, degrees Rankine.
P_1 = Suction pressure.
P_2 = Discharge pressure.
R = Gas constant, 53.4 for air.

Example: Estimate the power required to drive a reciprocating compressor capable of compressing 5000 cu. ft. "free" air per min. to 150 psi. gage from suction state of 14 psi. abs. and 75° F. Compression efficiency 92%, mechanical efficiency 90%.

"Free" air is air measured under standard atmospheric conditions. The discharge in pounds is obtained by application of the general gas law.

$$w = \frac{14.7 \times 144 \times 5000}{53.4 \times (460 + 60)} = 382 \text{ lbs. per min.}$$

Air under moderate temperatures has a constant z of 0.286.

$$\text{Ideal } W = \frac{53.4 \times (460 + 75)}{.286}\left[\left(\frac{164.7}{14}\right)^{.286} - 1\right] = 102{,}000 \text{ ft. lbs. per lb. air.}$$

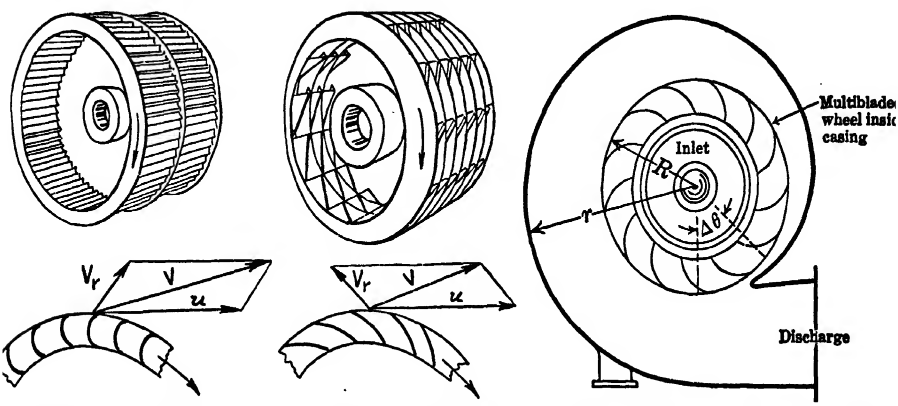
$$\text{Actual } W = \frac{102{,}000}{.92 \times .90} = 123{,}200 \text{ ft. lbs. per lb. air.}$$

$$\text{Horsepower} = \frac{123{,}200 \times 382}{33{,}000} = 1425 \text{ hp.}$$

6–5. Fans. Centrifugal action whereby a gas is compressed has been described heretofore. When large volumes are to receive a small compression (pressure rises of 1 lb. per sq. in. or less) the device is called a *fan*. Centrifugal fans fall in this category and fulfill a variety of needs such as ventilating, heating, combustion draft, and drying service.

In contrast to the high-pressure centrifugal compressor, the fan has narrow blades with very little compression occurring in the blading. However, the blade action on the gas is to increase its speed, thus requiring a diffusion to gain pressure. This diffusion is accomplished in a scroll case surrounding the wheel and comprising the casing of the fan. Diffuser guide vanes may sometimes be inserted in the scroll case to improve efficiency by reducing turbulence. Simple radial blading is sometimes used on account of its cheapness and simplicity, but most fans have blading that is curved. Figure 6–9 shows the appearance of wheels incorporating in the one case forwardly, and in the other, backwardly curved blading. For the same relative velocity of the

gas leaving the wheel and the same wheel speed, the absolute velocity is greater with forwardly curved blades. Achievement of efficient diffusion is therefore more important with forwardly curved blades and pressure increase is greater. Conversely for same pressure increase, forwardly curved blading may operate at lower rim speeds. However, backwardly curved blading can be built so that somewhere near the best operating point (maximum efficiency) pressure rise diminishes more rapidly than volume increase and thereby induces a self-limiting feature in power consumption that is desirable in many



Forwardly curved blades. Backwardly curved blades. Scroll case. Typically $r = R\left(1 + \frac{\theta^\circ}{300}\right)$

FIG. 6–9. Fan details.

applications. Power demand continues to increase with discharge in forwardly curved blading until well past the best operating point.

The mechanically driven draft fan is often used to supplement or supersede chimney action in the production of draft. Draft fans are designated as plate (paddle wheel), multivane, or propeller type, the latter being but seldom used for this service. The multivane centrifugal fan is the most common type.

Propeller fans operate with flow of air parallel to the axis of rotation. They may be grouped into (1) fans with thin sheet blades of metal or composition material, and (2) fans with blades of airfoil section. Examples of the first class are the table fans (the common domestic "electric fan"), small ventilating fans, ceiling fans, unit heater fans, and radiator cooling fans. With the exception of certain low-speed ceiling fans, these are characterized by high rotative speed and low efficiency. The blades are often stamped in a single piece from a sheet of metal then twisted slightly and mounted on the motor shaft. These fans are employed to move air but are not satisfactory if the air is to be forced against an appreciable pressure increment. The second class of propeller fans reflects a more scientific application of the axial flow principle. Whereas the first class moves the air largely by impulse against the face of the fan blades, the airfoil sections of the latter class perform in accordance

with airfoil theory of lift. While more expensive, they are also more efficient, although the latter advantage is gained in sizes more suitable to industrial than domestic usage. The axial flow fan can move large volumes against light static pressures and is frequently more compact and more readily applied than the centrifugal type.

To investigate the power needed for fan drive, it will be necessary to consider "velocity head" as well as static pressure, since velocities are usually high and static pressure increments small. The total pressure head $\mathscr{P}$ is found by multiplying the $\mathscr{H}$ equation, page 88, through by ρg and neglecting any y term.

$$\text{Pressure head } \mathscr{P} = P + \frac{\rho v^2}{2}.$$

The change of $\mathscr{P}$, created by a fan is

$$\Delta\mathscr{P} = (P_2 - P_1) + \frac{\rho}{2}(v_2^2 - v_1^2).$$

When fans take their suction from ambient air, $P_2 - P_1$ is the discharge plenum, which call ΔP, and, v_1 being zero, $v_2 = v$, the discharge velocity. For this case, then,

$$\Delta\mathscr{P} = \Delta P + \frac{\rho v^2}{2}.$$

Fans are used primarily to move air, not to compress it. Since pressure changes are small, an excellent approximation which simplifies the energy treatment is to assume a constant volume of flow through the fan. Treating the air as incompressible, power absorbed by the air is flow volume multiplied by pressure increase. Hence

$$\text{Air horsepower} = \frac{\Delta\mathscr{P} V}{33{,}000},$$

in which $\Delta\mathscr{P}$ = Increase of total fluid head, pounds per square foot.
V = Volume of fan discharge, cubic feet per minute.

The mechanical efficiency of a fan is the ratio of the above theoretical power to the required drive power. Multivane centrifugal fans will usually exhibit an efficiency of from 70% to 80% at their optimum point, with radial plate type fans being somewhat poorer in performance.

6–6. Wind Energy. Since only slight horizontal variations in the density of the air can occur ordinarily, and the total volume of the atmosphere is substantially constant, it is evident that a wind is necessarily a circulation; that is, any movement of the air in one direction must be offset by a return current elsewhere. All such movements are the result of thermal convection due to the heat of the sun.

The winds may be roughly classified into three types: (1) Winds which are in a sense permanent, including the easterly trade winds of the tropics and the westerly winds prevailing in the temperate zones. (2) Seasonal winds, such as the monsoons of the Indian ocean. (3) Local winds and storms, which temporarily interrupt the more general air movements prevailing at the time. Among these are the great cyclones and anticyclones characteristic of North American climate, ocean cyclones, thunderstorms, chinooks, tornadoes, and the very cold mistrals, and blizzards. The winds are practically all confined to a very few miles vertically above the earth's surface. Above this lies the stratosphere, believed to be a region of almost perpetual calm.

Windmill. Any arrangement of sails or blades which will turn the shaft to which they are attached when exposed to a wind may be called a windmill. The practical use of the windmill is to obtain for local use some of the energy represented by atmospheric motion.

There are four well-differentiated types of windmills, viz.:

1. The multibladed turbine wheel, or American type.
2. The Dutch type.
3. The propeller type high-speed wheel.
4. The rotor.

Although its large four-sailed wheel is efficient, the Dutch type, being difficult to regulate and operate, and having a higher first cost, has been used mainly in its place of origin, the Low Countries of Europe. The rpm. is lower than the American wheel, but the tip speed is relatively higher because of the length of the sails. Regulation is accomplished by turning the movable top of the tower into the wind.

Turbine windmills for pumping have become standardized in form and design and are usually rated in terms of the gallons per hour they will lift through various heights. The design of the wind wheel for electric generation is susceptible of much greater perfection since the starting torque is small and the running torque is uniform throughout the revolution, resulting in constant wheel velocity in a wind of constant velocity.

The kinetic energy of a mass of air m moving with velocity v is, of course, $\frac{1}{2}mv^2$ ft. lbs. Power in the wind sweeping horizontally through an area, in the vertical plane, of A sq. ft. at v ft. per sec. is $(Avd/2g)v^2$ ft. lbs. per sec., in which d is the air density. Employing d as 0.0763 lb. per cu. ft., a wind horsepower equation can be deduced.

$$\text{Wind power} = \frac{Av^3}{146{,}000}\ \text{hp.},$$

in which A = Cross section of wind stream, sq. ft.
v = Wind velocity, miles per hr.

That the wind represents considerable potentially available power is shown by solving the above equation for an area 100 ft. wide by 100 ft. high, using a velocity of only 30 mi. per hr.

$$\text{Wind power} = \frac{100 \times 100 \times 30^3}{146{,}000} = 1850 \text{ hp.}$$

A windmill with a turbine wheel D ft. in diameter could be considered to have available the wind power passing an area of $\pi D^2/4$ sq. ft. The power actually developed by the machine will be but a fraction of the wind power—about 10% for the turbine type wheel, and 20% for the Dutch type.

6–7. Measurement of Air. Measurement of the quantity of a fluid so nearly imponderable as air is troublesome. Small quantities of gases (i.e., fuel gas) are measured in meters that operate by positive displacement and pass definite volumes of gas each stroke or revolution of the mechanism. But air measurement so often involves large quantities that volumes are generally measured by obtaining the velocity at which air is passing through an opening of known area. Then weight may be computed from the general gas law provided the temperature and pressure of the air at the measuring station are known.

Example 1: Warm air at 120° F and atmospheric pressure passes through a circular opening of 18 in. diameter at a velocity of 900 ft. per min. What weight of air passes the opening per minute?

The volume represented by these data is $\frac{\pi \times 18^2}{4 \times 144} \times 900 = 1590$ cu. ft. per min. By the use of the gas law $PV = wRT$, the weight is found as follows:

$$w = \frac{14.7 \times 144 \times 1590}{53.4 \times 580^\circ} = 109 \text{ lbs. per min.}$$

The above example implies a simple calculation of air weight, but difficulty may be experienced in obtaining accurate knowledge of the velocity. Experimental methods available are:

1. Nozzle or orifice. Air can be discharged to the atmosphere through a well-shaped nozzle from a tank where it is under a few pounds per sq. in. gage pressure. The equations which are used to calculate velocity contain expressions for the gage pressure, the nozzle area, air temperature, as well as a *velocity coefficient* which is actually only a multiplier to relate theoretical to actual velocity. The coefficient is "handbook data" and must be selected to fit the individual conditions. An orifice will do in place of a nozzle. It is easier to construct, but information on the value of the aforementioned coefficient is less exact than with nozzle flow. The discharge of compressors is often measured by throttling that discharge into a tank operating at a pressure

that is convenient and accurate for measurements with the nozzle. The standard calibrated nozzle for which the velocity coefficient is known is then set into the side of this tank so that the air can blow freely through it into the atmosphere.

2. Anemometer. This is primarily an instrument for measuring surface wind velocities, and although it is suitable for measurement in tunnels and other large conduits, it is impractical for ordinary duct work. Spokes radiating from a central hub have a hollow hemispherical cup fastened to the tip of each spoke. This whirligig arrangement when exposed to a wind rotates at a speed that is proportional to the wind velocity. It is furnished with a calibrated indicating instrument.

3. Pitot-piezometer (pitot-static) tube. The pitot tube has an opening turned upstream in a fluid flow. It receives the impact of the current against it, which, could it be completely converted into pressure head, would produce in the pitot tube a pressure head of $v^2/2g$, superimposed on the existing static pressure of the fluid. When the pitot tube is used in connection with a piezometer tube, which receives no impact, the static pressure may be subtracted from the total pressure given by the pitot tube, leaving a difference which is velocity head. This arrangement is frequently used for measuring the velocity of flow of water or air.

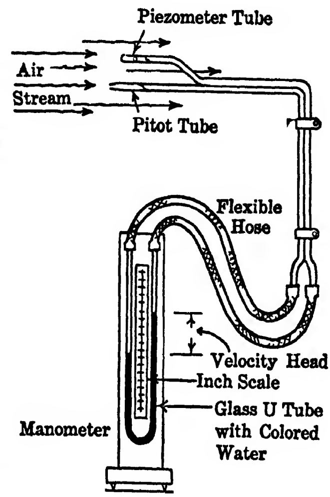


FIG. 6–10. Gas velocity by Pitot static tube.

A pitot-static head for measuring the velocity of air is here illustrated. The two leads from this head are carried to either leg of a U-tube manometer. When a current of air flows past the head the liquid is displaced in the manometer by an amount which is proportional to the velocity squared. When this manometer reading is converted into head of the fluid, i.e., air, the velocity is $v = C\sqrt{2gH}$. *

The reader must be careful not to misunderstand the meaning of H in this equation. It is not the height, y, read from the manometer (unless the fluid being measured is the same as the liquid in the manometer). In this equation H is the height of a column of the fluid sufficient to create a pressure equivalent to that set up by the fluid when it is efficiently brought to rest from a velocity of v. One converts the manometer reading into equivalent head of air by dividing it by the specific gravity of the air. This specific gravity is the air

* Note page 89.

density divided by the manometer liquid density. Air density at the existing pressure and temperature may be found from the general gas law by solving it for w/V.

Pitot-static measurements are an excellent way of determining velocities where air pressure is so nearly atmospheric that the nozzle method is unavailing, and where for other reasons (as small size of ducts) use of an anemometer is impractical. The coefficient C is more than .99 and can be assumed as 1.0 unless great precision of measurement is required. Since the pressure differential created by the tubes is extremely small, special sensitive manometers or differential pressure gages are used.

Example 2: A pitot-static tube is used to explore the air stream moving in a 24-in. × 32-in. duct. The air is under a slight vacuum, but this has hardly any effect on its density, so we will consider that it has the standard 60° F density of .0763 lb. per cu. ft. Although the air is moving faster at the center of the stream than near the edges, readings are taken at several stations, and it is determined that the average manometer reading, y, is 1.056 in. of water. If we want to find the flow in the duct, the procedure could be as follows: Convert y into equivalent height of a column of air, thus:

$$\text{Specific gravity of the air} = \frac{.0763}{62.5} = .001225.$$

$$H = \frac{\frac{1.056}{12}}{.001225} = 71.75 \text{ ft.}$$

Then

$$v = \sqrt{2 \times 32.2 \times 71.75} = 68 \text{ ft. per sec.}$$

$$\text{Flow} = Av = \frac{24 \times 32}{144} \times 68 = 363 \text{ cu. ft. per sec. } (21{,}800 \text{ cu. ft. per min.}).$$

6–8. Compressed Air Plant. An assembly of equipment for the purpose of obtaining a supply of compressed air is a "plant." It may range from the extreme simplicity of being little more than the compressor and its drive, to a complex assembly where, due to requirements for high pressure, and cool, clean, dry air, many other pieces of equipment are required. Many compressed air plants are required to be mobile—or at least portable—because of the extensive use of this medium for developing power in the construction field. Both air and water cooling of cylinders will be found in stationary plants but air cooling predominates in the portable types.

Advantages in compression can be secured by multi-staging the compression—that is, raising the pressure partially in a low-pressure cylinder, then completing the compression in a high-pressure cylinder. Not only are there mechanical advantages in having separate high- and low-pressure regions, but there are definite thermodynamic improvements both in energy and temperature control.

With two-stage compression, air is compressed to an intermediate pressure in the low-pressure cylinder, or cylinders, and cooled by an *intercooler* to

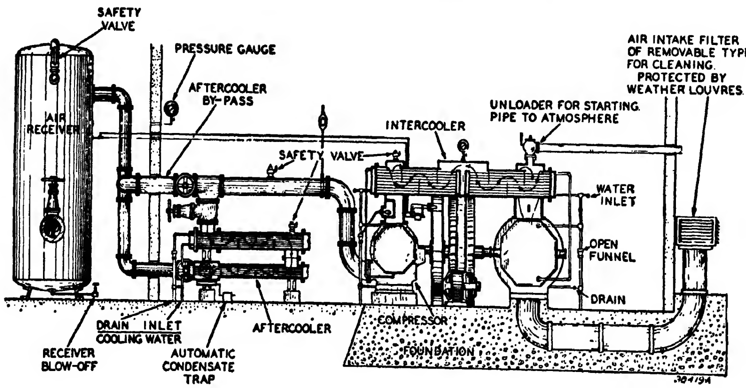


FIG. 6–11. Compressed air plant. (Courtesy Ingersoll Rand Co.)

approximately first stage intake temperature before it enters the high-pressure cylinder. In this manner the actual volume entering the high-pressure stage is less than the volume leaving the first stage, although, of course, its weight

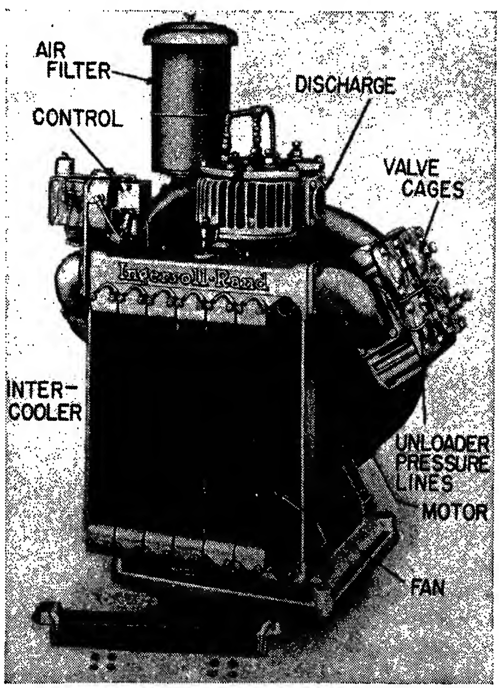


FIG. 6–12. Air-cooled, compounded compressor. (Courtesy Ingersoll Rand Co.)

is the same. The power that would be required to compress this difference in volume from the intermediate to the final discharge pressure is therefore saved.

There is also a saving in piston displacement required to handle a given quantity of free air per minute. Compared to the maximum temperature of compression of a single compressor, compounding lowers the final temperature through approximately the same range that the intercooler produces. The intercooler is sometimes a shell and tube type of cooler with water circulating through the tubes to absorb the heat of compression. Air-cooled compressor plants usually have finned intercooler tubing so that air blown externally over the tubing will cool the compressed air flowing inside.

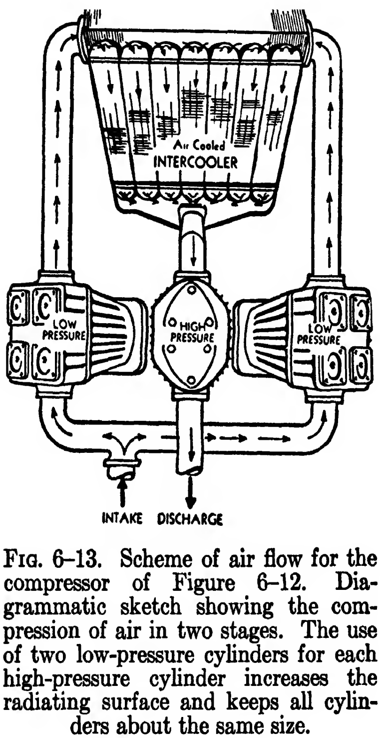


FIG. 6–13. Scheme of air flow for the compressor of Figure 6–12. Diagrammatic sketch showing the compression of air in two stages. The use of two low-pressure cylinders for each high-pressure cylinder increases the radiating surface and keeps all cylinders about the same size.

A large two-stage water-cooled compressed air plant is shown in Figure 6–11. Piping and some auxiliary equipment not mentioned in the text can be examined in this illustration. Such an installation is obviously of a permanent character. Of a more portable nature is the three-cylinder two-stage compressor and intercooler unit shown in Figure 6–12. Two cylinders which comprise the low-pressure stage divide the volume between them but discharge to the same intermediate pressure in the intercooler header. From the intercooler the partially compressed air is drawn into the intake of the central high-pressure cylinder where it is boosted to the final pressure, which in this design can range from 80 to 125 psi. gage. The cylinders are radial in arrangement, there being only one crankpin on the crankshaft. A slow-speed electric motor bolted to the compressor frame is direct-connected to the crankshaft. Hence belts and pulleys are eliminated, contributing to simplification of the unit. Cooling air is drawn through the intercooler by a fan.

Example: A two-stage compressor with first-stage piston displacement of 200 cu. ft. per min. is driven by a motor. Motor output is 35 hp. Suction temperature 70°, volumetric efficiency 85%, mechanical efficiency 95%. The intercooler pressure is 30 psi. gage. Air temperatures in and out of the intercooler are 220° and 110°. Final discharge pressure is 100 psi. gage, suction estimated 14.5 psi. These data will be analyzed to illustrate some of the principles heretofore mentioned.

$$\text{Adiabatic single stage work, } W = \frac{53.4 \times 520}{.286}\left[\left(\frac{114.7}{14.5}\right)^{.286} - 1\right]$$

$$= 79{,}600 \text{ ft. lbs. per lb. air.}$$

Weight handled = $200 \times .0763 \times 85\% = 12.95$ lbs. per min.

$$\text{Internal hp. per lb. air} = \frac{35 \times 95\%}{12.95} = 2.57 \text{ hp.}$$

$$\text{Compression efficiency * } \eta_c = \frac{79{,}600}{2.57 \times 33{,}000} = 93.8\%.$$

$$\text{Volumetric reduction by the intercooler} = \frac{wR}{P}(\Delta T)$$

$$= \frac{12.95 \times 53.4}{(30 + 14.7)144}(220 - 110) = 11.85 \text{ cu. ft. per min.}$$

$$\text{Power saving by intercooling} = \frac{11.85 \times 144(100 - 30)}{33{,}000} = 3.6 \text{ hp., approx.}$$

Heat removed in the intercooler, assuming $c_p = .24$, $Q_i = 12.95 \times .24(220 - 110)$

$= 332$ B.t.u. per min.

Compression and cooling of air reduce the quantity of water vapor it can hold. As all atmospheric air contains some moisture (represented by its *humidity*), it can be expected that intercooler shells and headers, receivers, and piping will accumulate water that was precipitated from the air because it reached a dew point while cooling. Such equipment should be equipped with drains so that such accumulations may be periodically blown out. Pipe lines have to be pitched so as to insure drainage of any of this precipitate that occurs in them.

Most compressed air equipment uses air intermittently. The discharge from the compressor must be correspondingly varied or the receiver pressure will increase until safety valves release the excess. The purpose of *control* is to regulate the compressor output automatically in accordance with the air requirements.

When the use of air is spasmodic, automatic start-and-stop control is generally used unless the unit is quite large, in which case the repeated starting cycles of the large motor are electrically undesirable. Starting and stopping is accomplished by a pressure switch, usually located on the air receiver. When air pressure drops a predetermined amount the compressor is started automatically. When the tank has been pumped up to a predetermined higher pressure the compressor automatically stops and remains shut down until usage of air again drops the pressure to the lower limit.

If air usage is relatively constant or normal demand approaches the compressor capacity, a constant speed control is preferable. Such controls are

* Based on isothermal compression $W = 53.4 \times 520 \log_e 114.7/14.5 = 57{,}100$ ft. lbs. per lb. air., and $\eta_c = 67.3\%$.

said to *unload* the compressor, meaning that as the upper pressure limit is approached the compression cycle is altered so that discharge is curtailed. Methods used to unload consist of (1) bringing into action enough of several cylinder head *clearance pockets,* (2) holding intake valves open, (3) throttling the intake. Constant speed control loads the compressor as the air demand continues and unloads it if the demand decreases momentarily. Although furnishing air only as required, the compressor continues to run unless shut down by hand. Sometimes both on-and-off and constant speed control are furnished so that while a fairly steady demand exists constant speed operation is possible, but the operator can switch over to start-and-stop control during periods of intermittent demand. This is called *dual control.*

6–9. Compressed Air in Use. The uses of compressed air in industry and construction are multitudinous. So varied are the applications of energy in this form that little is possible here beyond a general classification.

1. Direct use in which the objective is attained without the use of air-driven machines.
2. Air hoists, presses, and jacks.
3. Air-driven tools.
   Reciprocating action—free piston motion.
   Rotary action—turbines and positive displacement.
4. Miscellaneous. Leak testing, production gaging, etc.

Compressed air may be used directly in many ways. Examples are (1) the agitation of liquids by rising air bubbles released below the surface, (2) paint spraying, (3) deep sea diving, (4) various cleaning and blowing operations, (5) air lift water pumps.

Pneumatic presses, jacks, and vises are quiet and fast acting. A long plunger in a barrel produces a piston and cylinder action. With a large enough plunger and high enough air pressure, enormous forces may be produced at will. Hoists are generally operated by a geared air engine.

Compressed air tools represent the principal industrial use of air. A partial list would include such reciprocating compression tools as chipping and riveting hammers, rock drills, railway tie tampers, and various types of vibrators. Air motor tools are illustrated by drills, grinders, and assembly tools such as nut runners.

The reciprocating tools are most numerous in the percussion field, i.e., hammering, chiseling, riveting. Typically, a free piston in a cylinder is thrown back and forth violently by air pressure admitted alternately to its two faces. Rotary tools are usually powered by vane type motors which have, essentially, a reverse action from that of the machine shown in Figure 6–5. The high-speed motor will be geared down to the drive shaft. Table 6–1 shows typical air consumption of compressed air equipment.

TABLE 6–1. AIR CONSUMPTION TABLE

For Pneumatic Tools and Air Applications

These figures are approximate and are simply a rough guide for estimating the air required for desired applications, or the operations possible with a given amount of air available.

Tool or Operation	Air Consumption, Free Air (cu. ft. per min.)	Pressure Range	Tool or Operation	Air Consumption, Free Air (cu. ft. per min.)	Pressure Range
Chipping Hammers	11–37	60–100	Air Chucks	$\frac{1}{8}$–$\frac{1}{2}$ per operation	70–100
Riveting Hammers (Airplane Type)	8–16	70–100	Air Vises	$\frac{1}{8}$–$\frac{1}{2}$ per operation	70–100
Riveting Hammers (Medium)	21–40	70–100	Spray Painting	2.5–15	20–60
Rammers and Tampers	13–30	60–100	Fender Straighteners	4–10	60–100
Tie Tampers	8–17	60–100	Jackhammers (Small)	33–47	50–70
Pneumatic Diggers	22–63	60–100	Liquid Agitation	1–7 per sq. ft. of tank	Low
Drills	4–155	60–100			
Impact Wrenches	17–96	80–100	Wrenches	15–58	60–100
Nut Setters	7–17	60–100	Blowing and Cleaning	2–100	60–100
Screw Drivers	9–17	60–100	Door Operation	$\frac{1}{2}$–1 hp.	60
Grinders	7–80	60–100	Wood Borers	33–89	60–100
Wire Brushes	7–80	60–100	Paving Breakers (Medium Size)	18–29	50–70
Air Saws	27–45	60–100	Air Motors	4–390	60–100
Motor Hoists	$\frac{1}{3}$–25 cu. ft. per ft. of lift	60–100	Utility Hoists	$\frac{3}{4}$–3 cu. ft. per ft. of pull	60–100
Concrete Vibrators	22–100	60–100	Scaling Hammers	5–22	60–100

Courtesy Ingersoll Rand Co.

A typical compressed air tool is the flapper valve hammer shown sectionally in Figure 6–14. This tool might be used for chipping, caulking, beading, etc. Air for operation of the tool is brought through the handle and delivered to the chamber *F* above the valve. With the valve in the position shown by this illustration, air travels through port *D* and enters the cylinder at port *B*, then forces the piston to the top of the cylinder. During the first part of the

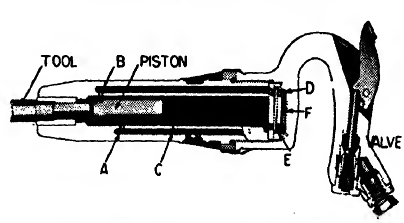


FIG. 6–14. Air hammer. (Courtesy Ingersoll Rand Co.)

stroke, the air above the piston is exhausted through port C. When the piston has passed this port, the remainder of the air is compressed in the top of the cylinder and throws the valve to open port E. Air can then no longer pass through port D, but must pass through port E, and again forces the piston toward the bottom of the cylinder. The air below the piston is exhausted first through port C and then through port A. However, when the piston passes port A, air is compressed below the piston and the pressure acting

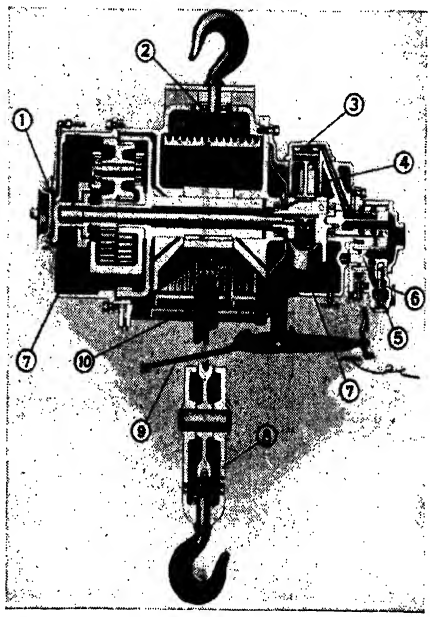


FIG. 6–15. Air hoist. (Courtesy Ingersoll Rand Co.)

through port B again opens the valve at D. This action is repeated continuously as long as air is admitted through the throttle in the handle.

The hoist shown in Figure 6–15 is motivated by a four-cylinder, single-acting radial air engine (3). Air admitted at the right is controlled by the throttling and reversing valve (5). The air is admitted to and released from the cylinders by a rotary distributor valve. Action is, of course, two-cycle. The engine crankshaft is extended to the gear case (7), where a planetary gear train reduces the speed to suit hoisting service.

Other rotary air motors will be found, acting on the rotating vane principle.

PROBLEMS

1. To what temperature would air rise in a *simple* compressor, i.e., not compounded, if it were given an adiabatic compression from standard atmosphere conditions to 60 psi. gage? Would isothermal compression have resulted in a higher or lower temperature?

2. What would the final temperature of the air mentioned in Problem 1 have been had the compression been isothermal? Would adiabatic compression have resulted in a higher or lower temperature?

3. If it were desired to hold down the temperature rise during a compression so that 200° F was not exceeded by a compression to 60 psi. gage, how much heat would have to be removed from each pound of air? (Hint: Use the data to compute n. Since

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n}, \quad \frac{n-1}{n} = \frac{\log (T_2/T_1)}{\log (P_2/P_1)}.$$

Then employ the general polytropic equation for transferred heat, Q, given in Chapter 1.)

4. Acetylene is being compressed by stages from atmospheric pressure and 90°. Between the first and second stages the pressure is 90 psi. and the gas is cooled from 300° F to 140° F. How much heat must the jackets of the first stage reciprocating compressor cylinder abstract per pound of acetylene? (See Problem 3.)

5. A reciprocating compressor raises air from atmospheric pressure to 100 psi. abs. atmospheric temperature 70° F. Consider that the volume of air in the cylinder at point a, Figure 6–3, is 1.0 cu. ft., that the compression exponent is 1.3, that the pressure ratio is 7, and that the clearance is 3%. Then plot the line abc to scale of 1 in. = 0.2 cu. ft. and 1 in. = 20 psi.

6. Repeat Problem 5 for a pressure ratio of 6 and discharge pressure of 84 psi.

7. An air compressor has point c, Figure 6–3, at 50 psi. gage, and line cd represented by $PV^{1.2} = C$. Clearance is 5%, and piston displacement 1.0 cu. ft. per stroke. With the compressor making 150 suction strokes per min., and the suction condition being 14 psi., 70° F, how much "free air" (i.e., air under standard conditions) is being handled by the compressor per minute?

8. A single-acting air compressor, 3 in. × 4 in. × 400 rpm., is to handle 5.0 cu. ft. free air per minute. Discharge at 85 psi. abs., suction conditions 14 psi., 60° F. Clearance air expands according to $PV^{1.2} = C$ (line cd, Figure 6–3). How much clearance can be allowed? Let x = clearance and set up an equation for volume da in terms of x.

9. Plot two graphs, each representing the compression of air in a reciprocating compressor from 14.7 psi., 1 cu. ft., to 100 psi. Clearance 0. One is to have isothermal compression; the other adiabatic. Shade the areas representing the respective effective work of compression and delivery. Scales 1 in. = 0.2 cu. ft., 1 in. = 20 psi.

10. Repeat Problem 9 for 120 psi. discharge pressure.

11. In a two-stage tandem compressor (tandem means both cylinders have the same stroke) the air is cooled from 300° to 150° in the intercooler after having been compressed to 80 psi. in the first stage. Initial suction conditions are 14 psi., 60° F. In what ratio do the diameters of the high- and low-pressure cylinders need to be? Assume zero clearances.

12. Repeat Problem 11 for a compressor in which, instead of a tandem arrangement, we have each cylinder with its individual crankpin, but each cylinder is "square," meaning that bore = stroke.

13. Assume that the actual pressure ratio in a single-stage centrifugal compressor is 60% of the theoretical. Then what shaft speed (rpm.) would be necessary to compress standard atmospheric air to 22 in. mercury gage pressure? Impeller diameter 18 in. The theoretical speed corresponds to an assumption of $\eta_c = 1.0$.

14. An airplane engine has an inbuilt supercharger which is a single-stage centrifugal compressor having 11-in. diameter impeller. Gears connect the impeller to the crankshaft in the ratio of 7.15:1. When the engine is making 2325 rpm. the abs.

pressure of the discharge is 28 in. Hg. when suction is 16.9 in. Hg. abs., and suction temperature is 10° F. Compare this pressure ratio with that obtained by formula, using 82% compressor efficiency.

15. Assume that you are to specify the electric motor to power a reciprocating compressor with a capacity of 80 cu. ft. "free air" per min. to 100 psi. gage at 400 rpm. Compressor is equipped with 27-in. diameter V-belt pulley. You have been supplied with an estimated compression efficiency (80%) and mechanical efficiency, including belt (88%). Using standard conditions as the suction state, compute the necessary hp. and pulley diameter of an 1800 rpm. electric motor.

16. During a compressor test, the discharge was measured and found to be equivalent to 3.62 lbs. of air per min. Suction conditions 14.5 psi., 65° F, discharge 75 psi. gage. The compressor was driven by a three-phase, 220-volt A-C motor whose line current (per phase) averaged 30 amperes during the test. The power factor of this motor can be assumed to be 80%. Determine the following in the order stated:

a. Ideal compression work per minute per pound air per minute.
b. Ideal compression horsepower.
c. Input to motor, kilowatts ($=\sqrt{3}EIf$ watts).
d. Output from motor, horsepower. Assume motor efficiency of 82%.
e. Compressor efficiency ($\eta_c e_m$).

17. A certain centrifugal fan moves air into a square duct at the rate of 1500 cu. ft. per min., the duct being 10 in. × 10 in. in cross section. Static pressure in the duct is 3 in. water gage. Temperature 60° F. Calculate the air horsepower.

18. A ventilating fan supplies a building with 25,000 cu. ft. fresh air per min. at 40° F. What power is needed to drive it, given its mechanical efficiency = 75%. Discharge plenum 4 in. water, velocity 900 ft. per min.

19. An American type windmill with an aerodynamic efficiency of 10% is wanted to pump 10 gals. of water a minute from a well to a tank located 60 ft. above the water surface of the well, when the wind has a velocity of 20 mi. per hr. Consider that the friction losses of the system divert 30% of the energy received by the wheel. What wheel diameter is needed?

20. A high-speed propeller-type windmill is used to drive an electric generator for charging storage batteries. Considering that in a wind of 30 mi. per hr. it absorbs 32% of the energy of the wind passing its disk, which is 6 ft. in diameter, estimate the generator output. Combined mechanical and electrical efficiency of the unit, 70%.

21. One of the standard orifice formulas for gases is

$$\text{Flow} = CA_2\sqrt{\frac{1}{1-m^4}}\sqrt{\frac{2g(P_1-P_2)}{d}}.$$

m is the ratio of the diameter of the orifice to that of the circular duct in which it is located. For a rectangular duct and orifice, $m = \sqrt{\text{ratio of areas}}$. d is the gas density. The constant C varies with m, but is approximately 62%. Calculate the flow indicated by the following measurements taken on the flow of air through an orifice in a rectangular duct. Duct 12 in. × 24 in., orifice 6 in. × 12 in. Pressure differential at the orifice is 2 in. water, density .076 lb. per cu. ft. Compute the quantity flowing, cubic feet per minute.

22. Using the data of Problem 17, estimate the probable velocity head reading on a water manometer attached to a pitot-static tube turned upstream in this duct.

23. The air velocity in the jet of a wind tunnel is being measured by a pitot-static tube attached to a sensitive water manometer. What velocity is indicated when the manometer reads 2.227 in.? Temperature of the air in the tunnel 82° F, pressure 14.7 psi.

24. The following data apply to a compressor similar to that shown in Figure 6–12. Discharge pressure 100 psi. gage. Motor rating 15 hp., speed 870 rpm. Low-pressure cylinders 5 in. diameter, high-pressure 4 in. diameter. Stroke 4 in. Assume a volumetric efficiency of 85%, and mechanical efficiency 95%. Intermediate pressure 35 psi. gage.

a. Calculate the delivery in pounds per minute.
b. Find the adiabatic single-stage input horsepower.
c. Find the intercooler inlet temperature, assuming $n = 1.3$, p_1 14.7 psi., t_1 60° F.

25. Data of Problem 24.

a. Assume that the answer to part (a) is 5.12 lbs. per min., and that the high-pressure cylinder also has a volumetric efficiency of 85%. What must the intercooler outlet temperature be for the 4-in. high-pressure cylinder to carry this air?
b. If the intercooler inlet temperature is 230° F, how much compression work is saved by intercooling? How much heat is removed per minute?

CHAPTER 7

Refrigeration and Air Conditioning

7–1. Heat Pump. A power cycle takes heat at a high temperature, converts some of it into mechanical work, and rejects the remainder at some lower temperature. The machine in which the conversion occurs is called a *prime mover.* The refrigeration cycle acts oppositely. A refrigeration cycle takes heat at a lower temperature and rejects it at a higher. To do this, the cycle must receive a power input from an external source. The heat rejected is more than that taken in by the amount of work required to motivate the cycle. Theoretically, any reversible power cycle could create a refrigeration cycle. Actually, practical considerations cause modification of the reversed power cycle for refrigeration use. Nevertheless, the ordinary vapor compression refrigerating cycle resembles a reversed power cycle. In the power cycle work is done by an engine, and in the refrigerating cycle work must be supplied to a reversed engine, which then becomes a pump to elevate heat from a lower to a higher temperature. This *heat pump* is a compressor. Most refrigeration cycles make use of a vaporous refrigerant. In a power cycle the efficiency is the work done divided by the heat supplied. In the refrigeration cycle the efficiency expression is replaced by "coefficient of performance." A *coefficient of performance* is the heat energy extracted at the low temperature divided by the work which must be supplied to operate the cycle. It measures the cycle performance in that it is an expression of the refrigeration obtained per unit of work supplied to the cycle.

$$\text{Coefficient of performance} = \frac{JQ}{W},$$

in which Q = Heat in B.t.u. absorbed from the ice tank, cold storage room, etc., per pound refrigerant.

W = Work in foot-pounds supplied to the compressor per pound refrigerant.

The unit of refrigeration corresponding to the absorption of heat equivalent to the production of 1 ton of ice per day of 24 hrs. is called a "ton." Normally, about 144 B.t.u. must be abstracted to convert a pound of water to ice. Using this value, together with the proper conversion factors, a ton of refrigeration is found to be equivalent to a heat abstraction of 200 B.t.u. per min. This rate

of heat flow is 4.715 hp. As the coefficient of performance measures the ratio of heat abstraction to input, the power which would theoretically be necessary per ton of refrigeration capacity is

$$\text{Power} = \frac{4.715}{\text{coefficient of performance}}\ \text{hp.}$$

Example 1: Suppose we want to absorb 12,000 B.t.u. per hour from a certain region in which the temperature is 35° F by absorbing it as latent heat while evaporating liquid ammonia inside refrigerating coils. A diagram of this situation is shown in Figure 7–1. To find the quantity and state required of the liquid ammonia, it is necessary first to assume a difference in temperature between the cold region and the evaporating ammonia. Assume this is 5° F. Then the saturation temperature for the ammonia is 30° F. It is ascertained, from tables of the properties of ammonia, that the heat of evaporation at this pressure is 544.8 B.t.u. per lb. Therefore, if the ammonia vapor left the pipe coils exactly dry and saturated, 12,000/544.8, or 22 lbs. of the liquid ammonia should be supplied to the coils per hour. If the outgoing vapor were wet, more than 22 lbs. would be needed, whereas less than 22 lbs. would do if some superheat were added to the vapor. The possible degree of superheat is limited to the temperature difference which was assumed to be 5° in this example.

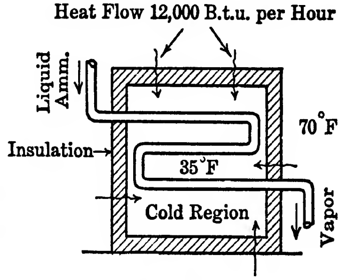


FIG. 7–1. Maintaining a cold region by absorbing the heat leakage into it in boiling ammonia.

Example 2: We now want to determine the coefficient of performance of the heat pump system which will take vapor from the coils and compress it at constant entropy until it reaches a pressure whose saturation temperature is as much as 90° F. Assume in this case that the state of the ammonia entering the coils is 88% liquid, 12% vapor, while leaving it is dry and saturated, both conditions at 30° F.

Tables show that, at 90° F, the saturation pressure is 180.6 psi. abs. To find the state of the vapor upon discharge from the heat pump, equate entering and leaving entropy. At 30° F, $s_g = 1.2790$. This must also be s at 180.6 psi., but not for a dry, saturated state. Inspection of the properties of superheated ammonia will reveal that at this pressure the entropy is 1.2790 at approximately 170° F. The enthalpy for the same state is found to be 687.3 B.t.u. per lb. Therefore, the increase of enthalpy caused by the heat pump = 687.3 − 620.5 = 66.8 B.t.u.per lb. 620.5 is h_g at 30° F. The refrigerating effect of the one pound of ammonia is the latent heat that the 88% that was liquid can absorb during evaporation. This is .88 × 544.8 = 478 B.t.u. per lb. The ideal C.P. is the refrigerating effect divided by the ideal work.

$$\text{C.P.} = \frac{478}{66.8} = 7.17.$$

7–2. Vapor Refrigeration. The general principle of vapor refrigeration is this. By evaporation, heat may be "loaded on" the working medium, and unloaded from it by condensation. The heat rejection through condensation may be made to occur at much higher temperatures than the heat absorption

if the vapor is compressed between evaporation and condensation. Thus, with the working fluid subjected to physical changes only, heat may be removed from one region and rejected to a region of higher temperature. That is *refrigeration.*

An elementary cycle of refrigeration can be operated with a minimum of four pieces of equipment. These are shown in an accompanying figure of a cold storage plant. Beginning the description with the refrigerating coils,

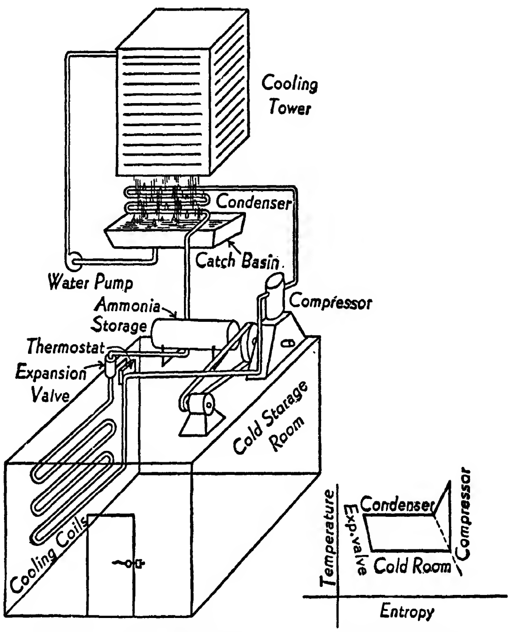


Diagram of refrigerating plant. Refrigerating cycle.

FIG. 7–2. Vapor refrigerating system.

note the bank of pipes in a cold storage room. In these pipes is the liquid whose boiling temperature is lower than the temperature of the cold storage room, so heat is passed into this liquid. It is absorbed as heat of vaporization, changing the liquid to a vapor. The vapor is withdrawn from the cooling coils by a motor-driven compressor which increases its pressure, until, at the discharge pressure of the compressor, the refrigerant has a saturation temperature high enough so that it may be condensed by some available cooling source such as water. The compressed vapor is discharged into a condenser. This is usually a bank of tubes over which cool water trickles. The heat of vaporization is passed from the vapor to the water, condensing the former to a liquid. The warmed water is either wasted, or cooled (by spray pond or cooling tower) and recirculated. The liquid refrigerant then is passed through an expansion valve to the lower pressure of the cooling coils. This is a throttling

process. The condition of the refrigerant as it emerges from the expansion valve is mainly liquid, a very small portion having been flashed into vapor. The delivery of the refrigerant from the cooling coil to the compressor takes place under the action of natural flow of a vapor in the direction of lowest pressure.

The vapor entering the compressor is dry, or nearly so, and compression raises it to a superheated state. This is shown on the cycle diagram. Known

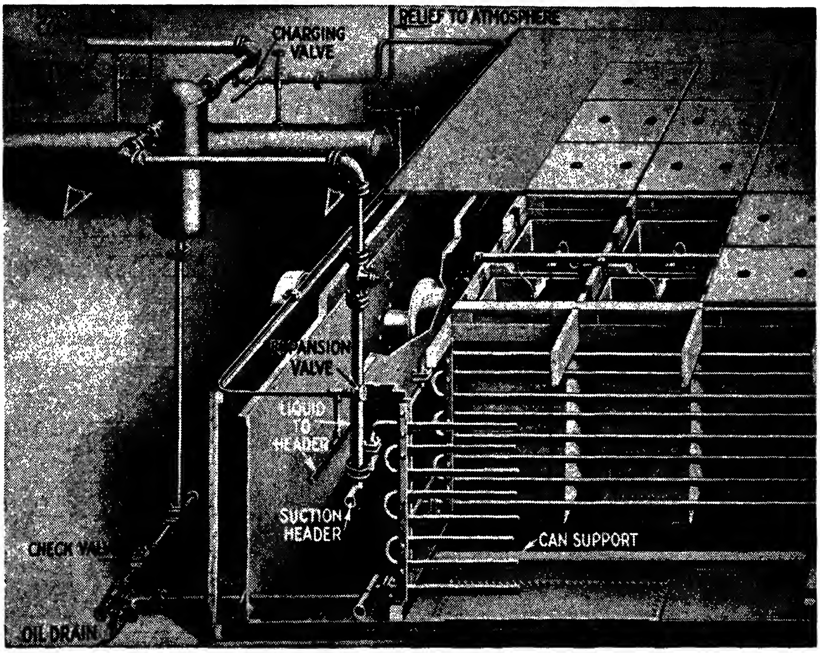


FIG. 7-3. Ice-plant, showing arrangement of coils and other details of the freezing tank. (Courtesy Frick Co., Inc.)

as dry compression, this entails considerably more work from the compressor than is needed merely to raise the vapor to the higher pressure. If the vapor could be brought back along the saturation line instead of the adiabatic line, a large amount of work would be saved, and the coefficient of performance increased. In practice this is sometimes attempted by:

1. Spraying a small amount of liquid refrigerant into the compressor cylinder (wet compression system).
2. Jacketing the compressor cylinder with cold water.
3. Cooling the refrigerant in coolers located between the cylinders of a multi-stage compressor.

Control of refrigerating of this system is exercised thermostatically on the expansion valve, allowing it to discharge more or less liquid ammonia into the

cooling coils. There are important differences between the system just described, which resembles that employed in central refrigeration or ice making plants, and the small domestic refrigerator. Whereas ammonia is the refrigerant chiefly used in the central plant, *domestic refrigerators* operate on a variety of refrigerants, including Freon, methyl chloride, and sulfur dioxide. Air is used as the condensing medium in a domestic refrigerator, and variable load operation is secured, not by operating the compressor at constant speed and controlling the expansion valve, but by a fixed setting of expansion valve and on-off operation of the compressor. The larger the refrigerating load the smaller the off intervals of time for the compressor motor. In one make, Freon vapor is compressed in a small cylinder by a piston whose connecting rod is driven from a crankshaft attached directly to the motor shaft. The compressed vapor is delivered at about 100 lbs. per sq. in. gage pressure to a condenser which resembles an automobile radiator. Air is drawn by a fan through the openings in the condenser core, and the vapor is reduced to a liquid. The liquid flows to a float chamber where a constant level is maintained by a float valve. As more liquid flows into this chamber, the valve passes it into the freezing unit, where it absorbs heat from the refrigerator box and is vaporized. The pressure there is approximately 20 lbs. per sq. in. gage. The vapor generated is drawn by suction into the compressor, where it is ready to begin the cycle again. Despite the small sizes, the coefficient of performance of the domestic refrigerators is not markedly inferior to the large-scale central plant.

Example: This example will show how to trace around an ideal vapor refrigeration cycle and determine the state of the fluid at key points. Referring to the cycle shown in Figure 7–2, consider the beginning of compression as state *a*, beginning of cooling, state *b*, beginning of expansion, state *c*, and beginning of evaporation, state *d*. Compression starts with the fluid dry and saturated. Quantities are for one pound of refrigerant. Again consider ammonia the refrigerant. Given temperature at *a* 10° F, at *c* 80° F. Tables are consulted when necessary.

State a: $t_a = 10°$ F, $p_a = 38.5$ psi., $v_a = 7.3$ cu. ft., $h_a = 614.9$ B.t.u., $s_a = 1.3157$.

State b: $s_b = 1.3157$, p_b, for condensation at 80° F, = 153 psi. At 153 psi. and $s = 1.3157$, by interpolation $h_b = 699$ B.t.u., and $t_b = 184°$.

State c: $t_c = 80°$, $p_c = 153$ psi., h_c is the enthalpy of saturated liquid, 132 B.t.u., likewise $s_c = .2749$ and $v_c = .02668$ cu. ft.

State d: $t_d = 10°$, $p_d = 38.5$ psi. Process *cd* is throttling from 153 psi. to 38.5 psi. With throttling, $h_d = h_c = 132$ B.t.u. Since the heat of liquid ammonia at *d* is only 53.8 B.t.u., the fluid must be partly evaporated at *d*. Note that $h_{fg} = 561.1$ B.t.u. Quality $= (132-53.8)/561.1 = 14\%$.

$$V_d = .0245 + .14 \times 7.280 = 1.045 \text{ cu. ft.}$$

$$s_d = .1208 + .14 \times 1.1949 = .2880.$$

7-3. Other Refrigeration. The vapor refrigeration just described might be specifically classified as the "mechanical compression vapor system." Although compression and condensation of a vapor in a closed cycle is the most common means of obtaining refrigeration, there are other systems. Usually these have some meritorious feature which recommends them for special conditions, but lack the general adaptability of compression refrigeration for a variety of applications.

Absorption System. A vapor may be dissolved in and thus absorbed by another liquid. An arrangement whereby the low-pressure refrigerant vapor

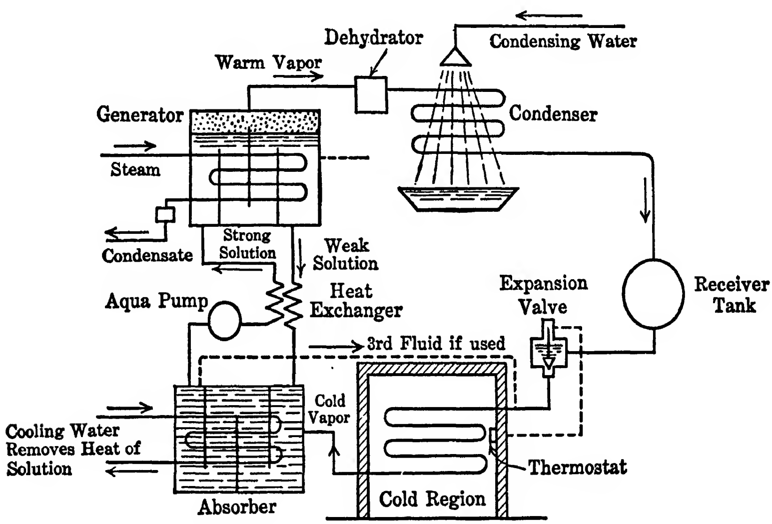


FIG. 7-4. Principle of the absorption system.

is dissolved in a carrying liquid, then pumped into a high-pressure chamber where the refrigerant vapor is evolved by heating is a means of eliminating the mechanical vapor compressor. Pressure rise is produced by a liquid pump for but a fraction of the mechanical work the compressor would require. The heat needed for evaporation of the high-pressure refrigerant vapor can be derived from exhaust steam or waste heat. The overall coefficient of performance based on energies is poor, but the input energy is of a cheap, low-grade form. This system could never be justified with high-grade input such as heating with electricity. A simplified flow diagram is shown in Figure 7-4. The refrigerant is ammonia, and the solvent water. Cool water readily takes into solution considerable quantity of ammonia, but will release it if subsequently heated. The refrigeration process from the refrigerant condenser back through the cold region is the same as for mechanical compression. At the absorber an inflow of a weak solution of water and ammonia (aqua ammonia) is mixed with the incoming vapor from the cold room coils, producing

a strong aqueous solution. The exothermic heat of solution is removed by cooling coils in the absorber. The strong solution is then pumped from the low-pressure absorber to the high-pressure generator, where it is heated by steam coils. This drives off a vapor which is principally ammonia, but contains a small quantity of water vapor. The weakened aqueous solution then flows back to the absorber, being cooled on the way by passing through a heat exchanger. Dehydration of the warm refrigerant vapor is necessary to prevent freezing in low-temperature regions due to water in the refrigerant.

A modification of this system eliminates the pump and expansion valve by use of an inert gas which circulates through what would otherwise be the low-pressure part of the system. The pressures are then substantially the same throughout and no pump and expansion valve are needed. Low refrigerant pressure required to achieve low temperature in the cold region is possible since in a mixture of inert gas and refrigerant vapor, the latter accounts for only part of the total pressure (see Dalton's Law, page 164). While this system is not well suited to large central plants, it is successfully practised in small units such as domestic refrigerators. The heat abstraction in this system can be by means of water, as shown in Figure 7–4, or by air. This method allows refrigeration where no electric service is available, using a gas or kerosene flame to heat the generator instead of the steam coil shown.

It is not practical to explain the absorption system quantitatively here because of the extensive empirical data required on the properties of aqua ammonia at various pressures and temperatures.

Steam Jet System. A plan for securing refrigeration using water vapor is sometimes used for producing chilled water for air conditioning and other cooling services where temperatures of the refrigerant do not need to be made lower than 40° F. Because steam jets are customarily employed to compress the vapor, the system is often designated as "jet refrigeration." Fundamentally, the medium is refrigerated by a process known as "flashing," a brief explanation of which is now in order.

Liquids may exist with thermal stability at high temperatures provided they are subjected to sufficiently high pressure. It is true of liquids in general that the lower the pressure on them the lower the boiling temperature and the lower the heat contained in the "saturated" liquid. Thus high-temperature liquids when passed from a region of pressure sufficient for stability into a low-pressure region may not be able to contain all the heat, originally possessed, as heat of fluid, and will be spontaneously partially evaporated by the surplus. This violent readjustment to thermal equilibrium is called "flashing," and is a common occurrence, having many uses but occasionally creating hazards. For example, the destructiveness of a boiler explosion arises mainly from the violence of flashing action, since the water originally contained in a ruptured boiler drum at 600 lbs. per sq. in. pressure suffers an al-

most instantaneous *four hundred fold* expansion in volume. The following example shows how this flashing, under control, can chill water.

Example: One pound of water at 60° F and atmospheric pressure is admitted to a region where the pressure is maintained at 0.25 in. Hg absolute pressure. What physical changes does the water undergo?

At 60° the water contained 28.06 B.t.u. per lb., whereas at 0.25 in. it can contain only 8.28 B.t.u. (both above 32° F). The excess goes into evaporation at the rate of 1070 B.t.u. per lb. steam, this being the latent heat at 0.25 in. Hg. Let x be the vapor produced by flashing. Equating the enthalpies before and after flashing,

$$28.06 = 8.28 + 1070x.$$

$$x = 0.0185 \text{ lb. vapor at } 0.25 \text{ in. Hg.}$$

$$1 - x = 0.9815 \text{ lb. water saturated at } 0.25 \text{ in. Hg.}$$

Since the saturation temperature of H_2O at 0.25 in. Hg. is 40.2° F, this flashing has produced almost a pound of chilled water at 40.2°. But notice that it has also produced .0185 × 2424 *, or 45 cu. ft. of steam to be removed from the high vacuum region in order to maintain a continuous process.

A jet system is pictured in Figure 7–5. The enormous volume of steam that has to be removed from the flash chamber cannot be economically

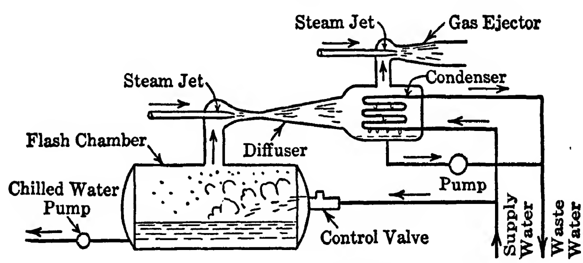


Fig. 7–5. Jet refrigeration.

handled by displacement apparatus. Steam jets are practical for this job, as they can compress these large volumes in jet equipment that is not unduly large. However, this enormous volume fortunately does not have to be raised to atmospheric pressure. If water is available at, say, 60° F, a compression to slightly over 0.5 in. Hg abs. would elevate the saturation temperature above 60° F so the vapor could be condensed by water coils and the condensate removed by a pump. Whereas it requires about 385,000 ft. lbs. to raise a pound of steam from 0.25 in. Hg to atmospheric pressure in simple adiabatic compression, it takes only 48,300 ft. lbs. to compress it to 0.7 in. Hg (68° F, sat. temperature), and *33* ft. lbs. to pump it out of the condenser as a liquid. Some non-condensable gases, released by flashing † from the supply water in

* Specific volume of steam at 0.25 in. Hg.

† The deaerating action of flashing is frequently useful, as in boiler feed water treatment.

which they were dissolved, will tend to accumulate in the condenser. They must be drawn out by another compression system. Steam jets are generally used for this compression also. Although compression here is to full atmospheric pressure, the energy involved is small because the volume of gas is moderately small.

7-4. Low Temperatures by Self-Cooling. In some cases the objective of chilling action is not to abstract heat from the surroundings, but to obtain

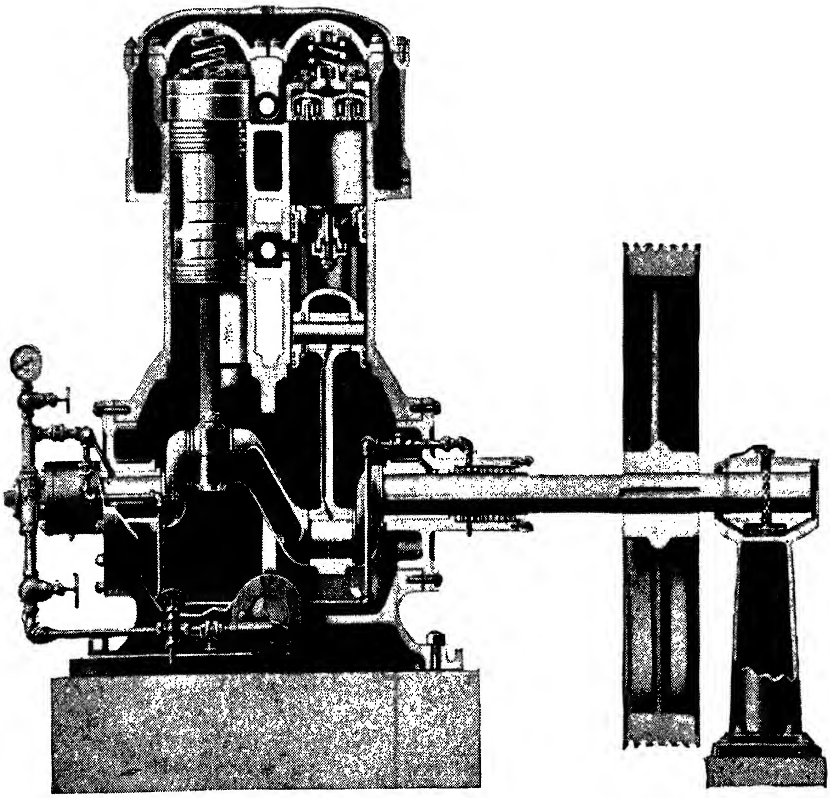


FIG. 7-6. Two-cylinder ammonia compressor. (Courtesy Frick Co., Inc.)

special thermal conditions in the medium itself. Thus, in the manufacture of dry ice (solid CO_2), and of liquid air, from which are derived liquid and gaseous oxygen, heat removal is sought for the purpose of creating a physical change in the working medium.

Dry ice is manufactured on a principle akin to flashing. The gas is first highly compressed in a mechanical compressor, then condensed to a liquid by heat exchange to water available in natural state. The high-pressure compression is necessary because of the high saturation pressure of CO_2 at normal atmospheric temperatures. When this liquid is flashed into a chamber at atmospheric pressure, most of it is instantaneously gasified, but some is

solidified into carbon dioxide snow. The latter is scraped together and pressed into the commercial product known as dry ice. The gas is withdrawn from the snow chamber, make-up CO_2 is added to it, and it is then sent back to the compressor. Commercial dry-ice production also involves staging and intertooling of compression, sub-cooling of liquid CO_2, and other practices designed to reduce mechanical work necessary per ton of CO_2 snow. The low temperature achieved is about $-110°$ F, the sublimation temperature of CO_2 at atmospheric pressure.

The production of liquid air is another process illustrating the attainment of very low temperatures through regenerative self-cooling. The common

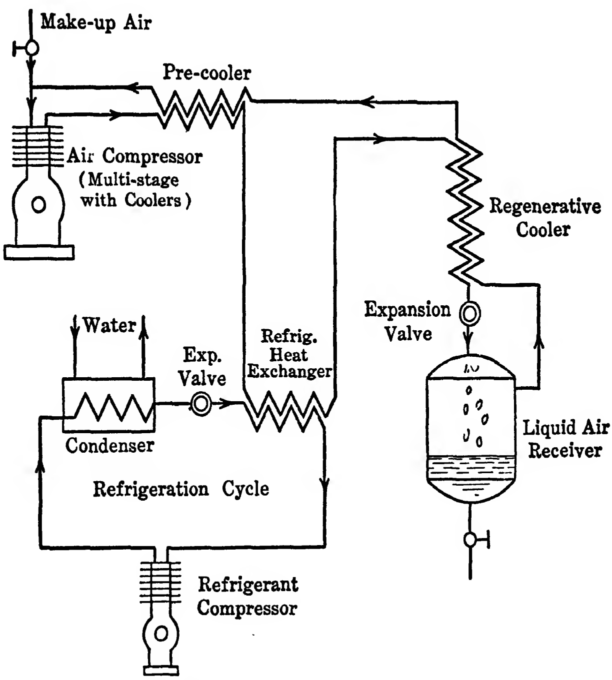


FIG. 7-7. Diagram of the liquid air cycle (Linde process).

method of liquefying air employed in this country represents a utilization of the *Joule-Thomson effect*, a brief discussion of which will follow.

The gas laws described in Chapters 1 and 2 are for ideal gases. The so-called permanent gases are not exactly ideal, although for much of engineering and scientific work they may be so considered. Sometimes their non-compliance with the ideal gas laws may be useful. The ability of the cycle shown in Figure 7-7 to produce liquid air is a case in point. The Joule-Thomson effect is represented in the cooling of a gas during throttling expansion. If an ideal gas were simply throttled to a lower pressure, the action should be isothermal, for if (as in all *throttling* processes) Δh is zero, then $wc_p\Delta T$ is zero. Obviously

ΔT is zero and the process is isothermal. But most actual gases gain slightly in enthalpy during isothermal expansion,* which property is equivalent to a $-\Delta T$ during a throttling expansion. If the gas were at its saturation temperature the change in enthalpy would produce partial condensation instead of further temperature drop. The cycle of operations implied in Figure 7–7 would, when flow was initiated, result in progressive cooling of the air in circulation with the heat being abstracted by the compressor inter- and after-coolers, and in the refrigerant condensing water. The liquid air receiver operates at about one atmosphere, so when the temperature drops to slightly below $-317°$ F, some of the air circulating (approximate magnitude of 5% to 8% by weight) will be liquified.

7–5. Humidity. It is appropriate now to mention a law associated with mixtures of two or more gases and/or vapors. The celebrated English chemist Dalton † (d. 1844) formulated the following principle which bears his name. *If several gases at the same temperature but not reacting chemically upon each other are introduced into the same container, the pressure of the resulting mixture is equal to the sum of the pressures which would be observed if each gas were separately enclosed in that container.* We may, for example, regard the atmospheric pressure as the sum of a nitrogen pressure, an oxygen pressure, an argon pressure, a carbon dioxide pressure, a water vapor pressure, etc. The same principle holds for mixtures of the saturated vapors of two or more liquids evaporating in the same closed space, provided one liquid does not dissolve the vapor from the other (as water dissolves ammonia). Like other gas laws, this law is approximately valid only within limits.

Another principle, derived from Avogadro's Law, should be stated here. At the same temperature and volume different gases have fluid pressures proportional to the number of molecules (or mols) present. Thus, if gases X and Y are mixed together in the proportion of M_x mols of X to M_y mols of Y, and together they have a mixture pressure of P, then since $P_x:P_y = M_x:M_y$, and $P = P_x + P_y$,

$$\frac{P_x}{P} = \frac{M_x}{M_x + M_y}.$$

P_x is the partial pressure of gas X. Although Avogadro's Law applied to gases, its application may include vapors without much error.

For many purposes the atmosphere may be considered as a mixture of a gas and a vapor. The gas is air with an average molecular weight of 28.8. The vapor is water vapor with a molecular weight of 18. Water vapor present in the atmosphere has a profound effect on several matters, such as human comfort, air conditioning loads, evaporative cooling, meteorology, etc. Quantitatively, the proportions are described by the *humidity*.

* No work done.

† Dalton is also well known for his pioneering in the atomic theory.

The most obvious way of expressing the absolute humidity or moisture content of the atmosphere is to state the percentage, by weight, of water vapor in its composition. For many purposes, a more useful quantity is the relative humidity, which expresses the vapor content as a fraction or percentage of the concentration necessary to render the vapor saturated at the given temperature. Specifically, the relative humidity of the air at any temperature is the ratio of the actual vapor pressure of the water vapor contained therein to the maximum or saturated vapor pressure of water vapor at the same temperature. At the dew point, the relative humidity is 100%. For most practical purposes the relative humidity also equals the moisture present in a sample of air, expressed as a percentage of the weight necessary to saturate that sample. A rise of temperature without the addition of more vapor reduces the relative humidity (but not the absolute humidity), while a fall of temperature increases it and may bring about saturation. Various forms of hygrometers have been devised to measure relative humidity.

The most common instrument is the so-called wet-and-dry-bulb thermometer, or psychrometer. Two exactly similar mercury thermometers are mounted side by side. The bulb of one is wrapped in a wick dipping into water, and is thereby kept wet. This bulb is cooled by the evaporation, the rate of which, and hence the resultant cooling, depends in a definite manner upon the relative humidity of the air.

The *dew point* is the temperature at which the air becomes "saturated" with water vapor, so that the relative humidity is 100%. It is the temperature at which the maximum saturation vapor pressure of water would be equal to the actual partial pressure of the water vapor in the atmosphere.

At the dew point, wet- and dry-bulb thermometers read the same. It is a simple matter to calculate the moisture present in saturated air, for the partial pressure of the vapor is the saturation pressure which may be taken from steam tables. Thus if we let X be water vapor and Y be air, then for a dew point of 40° F, p_x = .122 psi. The preceding equation can be set up as follows after slight rearrangement.

$$\frac{P}{P_x} = 1 + \frac{M_y}{M_x} = 1 + \frac{\dfrac{w_y}{28.8}}{\dfrac{w_x}{18}},$$

or

$$\frac{w_x}{w_y} = .625 \frac{P_x}{P - P_x} \text{ lb. vapor per lb. dry air.}$$

In this case

$$\frac{w_x}{w_y} = .625 \times \frac{.122}{14.7 - .122} = .00523 \text{ lb.}$$

When the air temperature is above the dew point, the vapor is present in a superheated state and its temperature does not directly indicate its pressure as when saturated. Wet-and-dry-bulb temperatures, along with certain of the thermodynamic properties of air and water vapor, can be used to calculate the composition and properties of the mixture, but such are beyond the scope of this book. Also there are available several specially constructed charts which allow an easy graphical solution of most all humidity problems, consequently thermodynamic calculations are seldom practiced in this field.

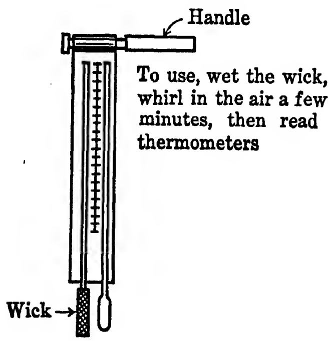


FIG. 7-8. Sling psychrometer.

7-6. Evaporative Cooling. The moisture contained in humid air is in the form of vapor, not liquid droplets. If a process occurs in which water is vaporized into the atmosphere, thus increasing its humidity, the proper latent heat of evaporation must be supplied from some source. Usually this source is the internal energy of the liquid itself, so what liquid remains after the humidifying moisture passes off is considerably chilled. This process is known as evaporative cooling, and has a variety of uses. It is a common personal experience that, when perspiring freely on a hot day, "one feels hot" if the relative humidity is high, and if no breeze is blowing. Lack of air motion, and high relative humidity both retard surface evaporation from the body, which thereby does not enjoy the full measure of evaporative cooling possible under more favorable atmospheric conditions. So, it is pleasant to feel the breeze of a fan—not because that air is cooler, but because of the improved evaporative cooling it engenders. Even air of nearly 100% humidity can produce evaporative cooling of warm water for, being warmed, its relative humidity decreases because the quantity of vapor necessary to saturate air increases with temperature.

Large quantities of warm water produced by energy conversions, and occurring in such fields as internal combustion engines, ice plants and steam plants, can be cooled and recirculated by transferring enough heat to the air by evaporative cooling. These devices are often called cooling towers. Their nature is shown in Figure 7-9, and again in Figure 7-12. The action that takes place is one whereby the stream of water to be cooled is broken up into a fine mist or rain, thus exposing a very large surface area. In this condition the water is mixed with air which is brought into the cooling tower by natural convection currents, or which is forced in by fans. If that air is not already saturated with water vapor, it will become so by coming in contact with the large area of moisture exposed in the cooling tower. The water is taken up by the air in the form of vapor at the partial pressure as determined by tempera-

ture. It will, however, be vaporized to steam before it can become part of the humidified air. The heat necessary to effect this evaporation comes largely from the water itself. Since to vaporize a pound of water requires some 1000 B.t.u., while to cool it 1° F requires only 1 B.t.u., it follows that an evaporation loss of 5% of the water passing through a cooling tower is capable of reducing the temperature of the remaining 95% some 50° F.

Cooling towers are constructed of wood or sheet iron having wooden, sheet iron, or terra cotta interior baffles so arranged as to convert the stream of water delivered to the tower at its top into a large surface of exposure. The cooled water s caught in a basin at the bottom of the tower.

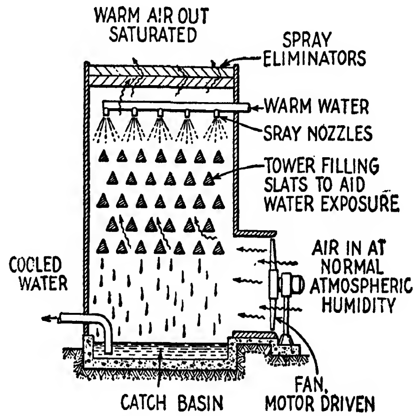


FIG. 7–9. Cooling tower.

7–7. Air Conditioning. This is the artificial treatment of air in buildings to render the living conditions of persons within the building more comfortable and healthful, or to ensure better conditions for the production and storage of materials. Complete air conditioning involves adjustment and control of the following operations performed on the air supply of a building:

1. Heating or cooling.
2. Humidification or dehumidification.
3. Ventilation.
4. Cleaning.

Although portions of this complete program of conditioning were often used in past years, it has only been recently that the importance of complete air conditioning has been fully understood. That a change of air in a room has beneficial effects is quite generally understood, and the ventilation of buildings has been a subject of considerable study. Although it is quite obvious that dust and obnoxious fumes have no place in either industrial or domestic rooms, few dwellings have filtered air supplies. One important fact which stimulated more scientific handling of the air-conditioning problem is that, in addition to temperature and air movement, the relative humidity is a very important factor in determining the comfort of the occupant of a room. Experimental research of the American Society of Heating and Ventilating Engineers indicated that air conditions which yielded equal degrees of warmth to human subjects plotted as straight lines on the common psychometric chart. These lines are called "comfort lines." The meaning of this is that a person

may feel equally comfortable at two different temperatures, provided the humidity of the air at these two temperatures differs by the proper amount. The reason for the influence of humidity on comfort is that the nearer to a completely saturated state the air in a room becomes, the less tendency there is for evaporation to take place from the body. This evaporation plays no small part in determining hot-weather comfort, and is a factor in surface cooling of the skin. On the other hand, in cold weather the atmosphere in an artificially heated room is very likely to prove deficient in moisture unless some special means for increasing the moisture content is provided.

Complete air conditioning, then, will involve the following equipment: a ventilating system for giving motion and circulation to the air; a furnace or heater to raise it to proper temperature in cold weather; a cooler to temper it for hot weather comfort; a humidifier, or dehumidifier; an air washer, or filter. This complete service is frequently applied to industrial buildings, theaters, auditoriums, etc., but has not been employed to any appreciable extent in homes. One reason for this, of course, is the cost of such equipment for the average home. Another is lack of information on the benefits realized from an air-conditioning installation; another, the widespread use of vapor or hot-water heating, which is not readily converted to an air-conditioning system.

7–8. Summer Conditioning. To show what thermal load a summer air-conditioning system must remove from a conditioned region, consider the case of a single room structure as shown in Figure 7–10. This shows the various ways by which heat gets into a region. There are others, such as hot food, hot water, etc., but generally cooling loads consist of: ① Conduction through walls, windows, roofs; ② direct radiation (sunload) through windows; ③ sensible heat brought in with ventilating air, as well as latent heat in the vapor it contains; ④ heat from incandescent lights; and ⑤ heat created by human metabolism. A person under average interior condition releases about 400 B.t.u. per hr., half as sensible heat, half as latent heat in the moisture exuded by the body and respiration.

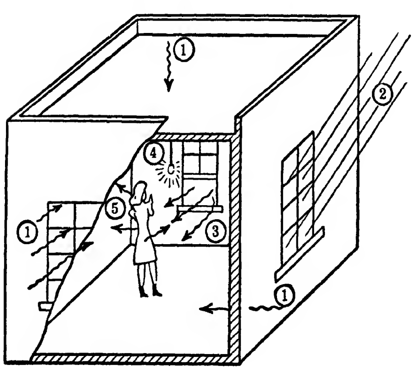


FIG. 7–10. Cooling load.

Heat transmission by conduction and convection was mentioned in Chapter 4. Research and accumulated statistics provide the data on coefficients used to determine the heat flow through walls, windows, roof, etc. The heat transfer coefficients are generally stated as conductances,* i.e., the B.t.u.

* Including convection in the adjacent air films, as well as conduction.

TABLE 7–1. HEAT TRANSFER FACTORS

The unit is B.t.u. per hr. per sq. ft. per deg. F temperature difference.

Outside Walls	
8-in. thick concrete	.70
12-in. thick concrete	.58
8-in. thick cinder block	.58
12-in. thick cinder block	.53
Conventional clapboard on wood studding with plaster interior finish	.24
8-in. brick, interior "furred" and plastered	.22
Partition Walls	
Plaster both sides on wood studding	.34
Insulating panels on wood studding	.12
4-in. cinder block, plastered both sides	.40
Ceilings	
Plaster on wood joists	.61
Plaster on joists, wood floor above	.24
Insulating panels on wood joists	.37
Roofs	
Built-up asphalt roofing on 4-in. concrete slab	.70
Same as above with addition of insulating panel, about $\frac{3}{4}$ in. thick	.30
Wood roof with asphalt roofing or shingles	.53
Same as above, with usual layer of insulation	.15
Wood roof with sheet metal covering or slate shingles	.60
Same as above, with usual layer of insulation	.20
Floors	
Conventional wood floor, plastered ceiling below	.24
Concrete slab on the ground	.30
Miscellaneous	
Windows *	1.00
Doors	.7
Glass blocks	.4

* Does not include direct radiant sunlight. This can be taken as 100 B.t.u. per hr. per sq. ft. of windows exposed to direct radiation.

transferred per square foot per hour per degree of temperature difference for a partition of stated composition.

These sources of heat may be present in varying degree, but represent an energy input that must be continuously extracted for maximum conditioned comfort. Objectives aimed at in designing conditioning systems are: (1) to maintain a fixed differential (10° or 15° F) between outside and inside temperature, with a minimum of about 75° F inside; (2) to maintain moderate

humidity, say about 50%, although most air conditioning systems allow this to fluctuate widely; and (3) to provide sufficient fresh air for respiratory comfort, generally not less than 3 cu. ft. per person per min., nor more than 25, but always as much as one complete air change in the region per hour. The elements of a summer air-conditioning system are shown in Figure 7–11.

Briefly describing this sytem, the conditioned space is supplied with cooled air through wall or ceiling outlets from a supply duct. The quantity and temperature of this air are adjusted so that when it has mixed with the room air

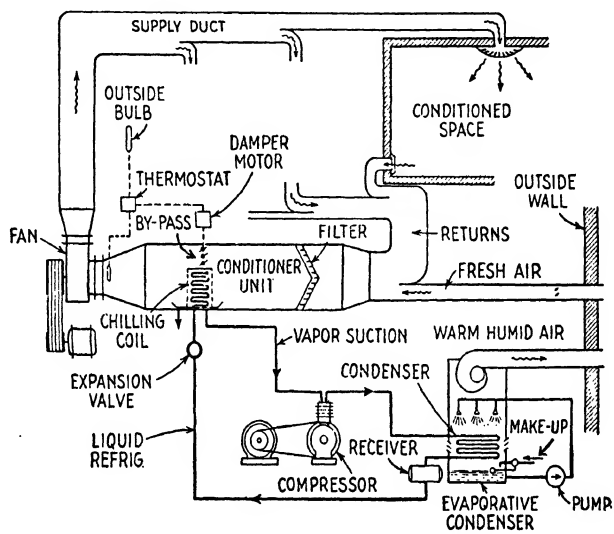


FIG. 7–11. Elements of a direct expansion air-conditioning system.

the temperature is depressed the desired amount below outside temperature. The difference between the duct air and the conditioned room temperature is called the *diffusion temperature.* It is an important characteristic of the conditioning system, for the larger it is the less the air required to be in circulation to pick up the heat leakage. But if it is made too great the occupants will experience chilly drafts. About a degree per foot of ceiling heights seems to fit existing practice. About three-quarters of the quantity supplied is taken into the return duct for recirculation; the other quarter is made up of fresh air intake for ventilation. Returns and fresh air are brought into the "conditioner unit" for cleaning, chilling, and dehumidification.* The air filter thickness has considerable effect on the quantity recirculated unless the fresh air is supplied from a fan. In the unit shown, the refrigerant is directly expanded to chilling coils, but some systems chill water by refrigerant expansion and circulate the water through the conditioner. Direct expansion is

* Winter conditioning can be incorporated by including heating coils and humidifying sprays in the unit.

cheaper, and well adapted to a central conditioner. Chilled water coils are desirable where several conditioners are used in scattered locations, being served from a central compressor. The design of the air conditioner and its controls varies greatly, depending on requirements as to precision of control of temperature and humidity sought; also on the ventilation requirements. In some cases the air may have to be overcooled in order to obtain sufficient dehumidification, then reheating coils must be used. The system shown would not give close control of humidity; however, average air conditioning

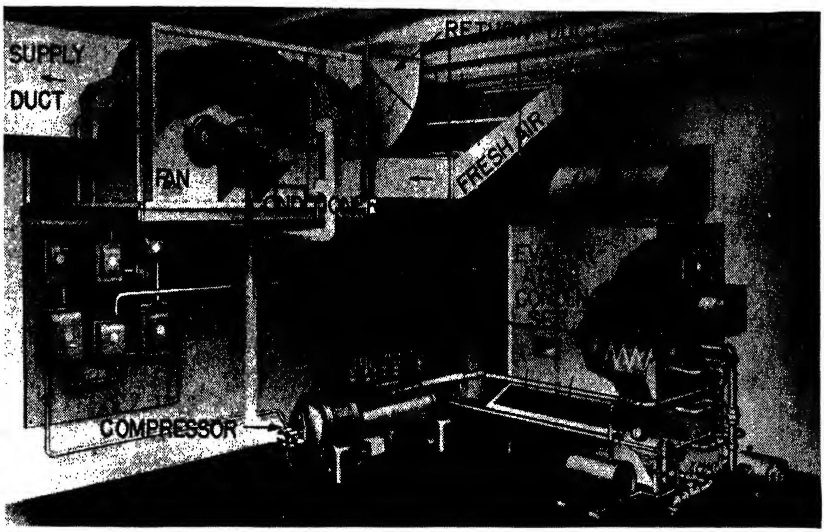


FIG. 7-12. Air-conditioning equipment. (Courtesy Westinghouse Electric Corp.)

does not require it. A by-pass damper causes more or less air to shunt around the chilling coil and so controls the air temperature in the supply duct. The expansion valve is set to maintain a given compressor suction pressure and variation of outside temperature is accommodated by varying the by-pass. This is secured by placing thermostat bulbs both in the duct and outside the building. Their joint effect is translated into a signal to the damper regulator increasing or decreasing the by-pass so that duct temperature will be $(t_d + t_c + t_s)$ below outside atmospheric temperature down to a certain fixed minimum, say 75°–80° F. In the above statement, t_d is diffusion temperature, t_c is desired temperature differential in the conditioned space, t_s is the temperature rise, if any, in the supply duct, including fan.

Example 1: What refrigerating capacity is required to condition an intermediate store floor 200 ft. × 80 ft., having a 12-ft. ceiling? Floors above and below are also air conditioned. 30% of wall area is glass, walls 8 in. brick, plastered inside. All four walls exposed. Maximum outside temperature 100° F, desired t_c 15° F. Maximum occupancy 320 persons. Allow for lights at 2 watts per sq. ft. floor area.

From Table 7–3 air change should be 2–5. Allowing two changes per hr. in a space having 200 × 80 × 12 cu. ft. is equivalent to 20 cu. ft. per person per min. This is quite adequate; in fact, 10 to 15 cu. ft. would be suitable, so we will reduce ventilating requirements to $1\frac{1}{2}$ changes. Net wall area = 70% (400 + 160) × 12 = 4700 sq. ft. Glass area = 2000 sq. ft. Maximum glass area passing sunlight at any time = 30% (200 × 12) = 720 sq. ft. The cooling load is summarized thus, using data from Table 7–1.

		B.t.u. per hr.
Heat transferred through walls, 4700 × .22 × 15°	=	15,500
Heat transferred through glass, 2000 × 1.0 × 15°	=	30,000
Radiant sunload, 720 × 100	=	72,000
Released by lights, 16,000 × $\frac{2}{1000}$ × 3412	=	10,900
Released by occupants, 320 (200 latent + 200 sensible)	=	128,000
Required to cool the fresh air $1\frac{1}{2}$ × (200 × 80 × 12) × 1.1 *	=	317,000
Total cooling load		573,400

Since 200 B.t.u. per min. is a "Ton" of refrigeration, the required capacity for this installation is 573,400/200 × 60 = 48 tons.

Example 2: Estimate the temperature and quantity of air to be circulated to the space described in Example 1.

Allow $t_d = 12°$, $t_s = 2°$, t_c, as in Example 1 = 15°.
Room temperature = 100 − 15 = 85° F.
Inlet grille temperature = 85 − 12 = 73° F.
Temperature at conditioner thermostatic bulb = 73 − 2 = 71° F.
Sensible heat to be picked up from conditioned space = 15,500 + 30,000 + 72,000 + 10,900 + 320 × 200 = 192,400 B.t.u. per hr.
The specific heat of air is taken as .0183 B.t.u. per cu. ft. per deg. F.
Let V = volume discharged into the region.
$12° \times .0183V = 192{,}400/60$; V = 14,600 cu. ft. per min.
The ventilation allowed was $1\frac{1}{2}$ × (200 × 80 × 12) = 4,800 cu. ft. per min.
In this case, ventilation is slightly over 30% of the air circulating.

7–9. Building Heating. Portions of buildings where persons work or live are heated when the outside temperature falls below that considered necessary for health and comfort. The heat that must be supplied equals the heat which is dissipated from the building. The amount needed varies with the difference in temperature between that maintained on the inside and that determined by the state of the weather. But other things also have a determining influence upon building heating—namely, the area and type of the exposed walls, the roof, the windows, and the leakage through cracks, ventilators, etc. Heat is lost from a building by conduction through walls, and by convection and radiation from outside surfaces such as windows, walls, roof. Heat is also dissipated by air leaking from a building, or by air intentionally discharged by a ventilating system.

Conductive heat transmission is calculated from conductance data, as in the case of summer cooling, but now the heat is passing outward through the

* Assumed B.t.u. required to cool and dehumidify 1 cu. ft. ventilating air.

walls. By multiplying the coefficient by the exposed surface and the difference between outside and inside temperature, part of the heat loss is determined. This, however, does not take care of heat loss by air leakage. One way to allow for that is to include a certain number of fresh-air changes per hour in the needed heating. For the ordinary room having windows, it is assumed that to allow for leakage, the heating system must supply heat to take care of a complete change of air each hour. (Halls, stores, factories, may need two or three times this allowance.) By computing the heat needed for each heated room, and totalling these for the building, the required heat output of a heating system may be determined. Whereas in summer conditioning, dehumidification is needed, the reverse is true during the heating season. Humidities in heated regions tend to be low, thus aiding evaporative cooling. Artificial humidification by sprays, etc., can result in equal comfort at lower temperatures than would be necessary in a dry zone.

A person gives off some heat, and although this is frequently disregarded as contributing anything to the heating, it will be necessary to reckon with this source of heat in any analysis of a public building, where large numbers of people may gather in one room. The building heating system and ventilation are jointly considered in a large building, especially a public one. With any attempt to insure adequate heating, the modern building owner usually will find it economical to spend money for heat insulation, since by proper coordination of investment in heat insulation, heating system, and fuel, a minimum annual heating cost can be secured.

Heat insulation is available commercially in a number of forms. The air space between studding, if blocked at the top floor levels, becomes a dead air space, and is effective in insulating against heat loss. Plaster and brick are fairly good conductors; wood, building paper, fiber wallboard, are heat insulators. Rock wool, either granular, fluffed, or in bats, is widely used where the maximum of heat insulation is attempted.

Central heating means the supply of heating service to a group of surrounding buildings from a central heating plant. The heat-carrying medium is sometimes steam, sometimes hot water. District heating is similar to central heating, but a distinction can be drawn as follows: A central system can be thought of as that which supplies a group of buildings which are under common building superintendence, or having common aim, as, for example, buildings of an educational institution, or a manufacturing plant. *District heating*, on the other hand, is a public utility service and applies to the heating of buildings in densely occupied city sections, from a public utility heating plant which sells heating service.

The instances where electric heating is of use are almost numberless in this day, since one may add to the large assortment of domestic electric devices, such as irons, corn poppers, percolators, etc., an equally great variety of com-

mercial devices, like annealing furnaces, bakers' ovens, glue pots, vulcanizers, etc. However, the use of electricity for building heating, although apparently successful in some experimental installations, is at a great disadvantage because of the high cost of electric heat compared to that obtained directly from the fuel.

7–10. Heating Systems. While the words "heating system" generally convey the idea of one of the indirect systems employed at the present time to heat homes or buildings, direct radiation is, itself, a system of heating. A stove located in the room which it is heating proves to be a very efficient means of getting heat from a fuel into the air of the room. As a system, it suffers from the following disadvantages:

1. Unsightliness.
2. Multiplicity of heating units required for a building containing a number of rooms.
3. Unequal distribution of heating.
4. Fire hazard.

Although most of the heat supplied by a stove is black body radiation, there is some amount of convection resulting from air currents sweeping up over the heated portions. A fireplace delivers no heat by convection; rather, it withdraws heat from a room by leakage up the flue. The heating effect of a fireplace comes entirely from radiation. As a heating system, a fireplace is inefficient and wasteful of fuel; however, this criticism must be tempered for those special fireplace designs (usually patented) having built-in convectors.

The adequate heating of buildings seems to be obtained best by the use of indirect heating, that is, generation of the heat at a central point, as a furnace, then loading that heat onto the medium which conveys it to the various rooms in the required amount. Release of that heat into the rooms is obtained by several methods, to wit:

1. *Radiators.* For transferring heat from water or steam directly to room air, a common device is to expose hollow cast-iron radiators to the room air and circulate the heating medium inside them. Transfer to the air is partly by radiation, but mostly by convection to the up-sweeping air currents. The design of radiators follows a fairly standardized pattern. As Figure 7–13 illustrates, a radiator unit is assembled as a number of sections, each of which has the same number of cast vertical cores, or tubes. Sections are available with as few as three, and as many as seven tubes; also in varying heights. Many different combinations of these produce heating surface in a variety of capacities and shapes, making it possible to select a radiator fitting the heating needs of a given space very exactly. If the radiator is exposed it is "free standing," as contrasted with one enclosed with a pressed metal case which assists the production of convection currents. Sometimes radiators initially

free standing have later been enclosed, but originally enclosed designs are generally worked out with the use of convectors.

2. *Convectors.* Free-standing radiators are not the most efficient heat transfer surfaces to be imagined, because a compromise must be struck between heat transfer and appearance. A unit designed to be enclosed, and thus

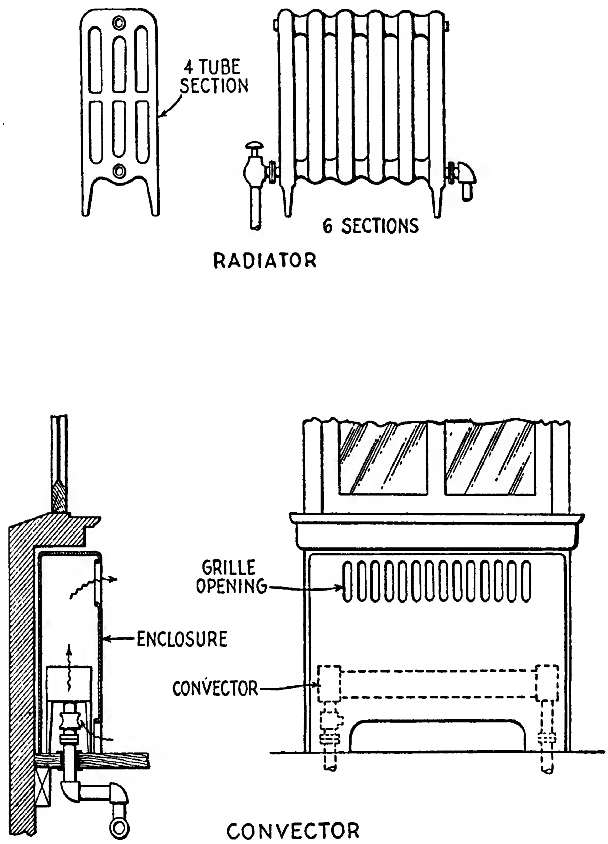


Fig. 7–13. Heat transfer surfaces in building heating.

hidden, can be designed for heat transfer alone. Such a unit is called a convector, and is always used with an enclosure. Enclosures of pressed steel are made in a variety of designs to meet architectural requirements. The one shown in Figure 7–13 is recessed in the wall under a window, a common arrangement. Cool air from floor level enters at the bottom of the enclosure and, being heated by contact with the convector, rises and discharges into the room through the open grillwork on the face of the enclosure. The convectors themselves are made of finned cast iron sections, or small copper tubes connected between headers. Both radiators and convectors are rated on the basis of a fictitious equivalent direct radiation (EDR) surface. If for steam,

a square foot EDR represents the ability to transfer 240 B.t.u. per hr. to the surrounding air, the steam temperature for this rating is 215°, corresponding to ordinary steam systems. A square foot transfers less heat when used with hot water because the interior temperature is lower than with steam. Based on 180° water temperature a square foot EDR transfers 150 B.t.u. per hr.

3. *Grilles and Registers.* The discharge of warm air, which has previously been heated by a central furnace into a room, is made to take place through register outlets. Leader ducts convey the warm air horizontally from the central warm air heating furnace to stacks, which are vertical ducts located in walls and partitions. Register boxes terminate the stacks and provide for the entry of the warm air into the room through registers which are the finish grillwork covering the opening where the register boxes terminate.

4. *Panel Heating.* Warmed panels in the walls, floor, or ceiling can radiate heat to the region they partially enclose. While electric heating has been experimented with, this is ordinarily accomplished with pipe coils carrying warm water. In panel heating large areas of mildly heated surface replace concentrated areas of high temperature emission, such as radiators. Where the pipe coils can be embedded in the floor or wall—as in concrete—panel heating is far better than in cases where an air space is left around the pipe. Warm water generated by a conventional heater is pumped through the coils, thus warming the whole panel.

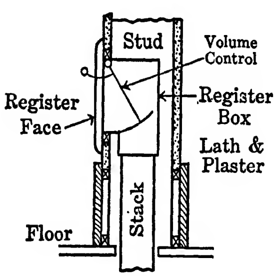


FIG. 7-14. Warm air outlet.

The different heating systems in use today may be classified on the basis of the heat-conveying medium, i.e., warm air, hot water, steam. Each of these systems is briefly described herewith.

In a *warm air heating system* the furnace is enclosed by a casing, leaving an air space between casing and furnace. From the casing ducts lead out to the different rooms. Air is supplied to the casing, either directly from the furnace room, from outside the building, or from return ducts which withdraw the air from the rooms. The air in the casing being in contact with the heated furnace, expands, becomes lighter than normal, and rises in the ducts until it is discharged in heated condition into the rooms through registers located in the floor or side wall. The older warm air systems had a circulation that was maintained entirely by the levity of the heated air leaving the furnace, but had certain disadvantages. For example, the effect of wind blowing against one side of the building caused infiltration which opposed duct air pressure in the windward rooms, an action which tended to give unequal distribution of heat, leaving the rooms on the windward side underheated, and those on the leeward side, overheated. Also, the furnace had to be centrally located, the

basement was encumbered with large numbers of ducts, and homes which were not roughly cubical in shape were not well adapted to this style of heating.

Recent developments in the field of warm air heating have led to the employment of forced circulation of the hot air by a motor-driven blower. Having this, it is not necessary to be so careful about friction losses, and smaller ducts, trunk-line systems, and filters are possible. Since the available pressure is of much higher order than that obtained from gravity alone, there is no difficulty about forcing air equally into all rooms. This system of heating, furthermore, is admirably adapted to air conditioning requirements. The operation of a forced circulation warm air heating system is controlled by a thermostat located in a representative room. The thermostat is connected electrically to a small draft-regulating motor, and in effect becomes a switch for that motor. A certain differential of a few degrees is allowed in a room. When the temperature sinks to a predetermined point, the thermostat operates the draft-controlling motor, which in turn closes the check damper and opens the draft so that the rate of combustion is increased. The temperature of the air in the bonnet of the casing rises until a thermostatically operated switch located there starts the blower. The warm air is then circulated in the rooms until the temperature rises to a predetermined limit, upon which the room thermostat regulates the draft. Soon the slower rate of combustion results in a cooler furnace bonnet, and the blower is thermostatically stopped. This cycle is repeated as frequently as needed, but the room and bonnet thermostatic switching do not necessarily stay in synchronism.

Hot water heating systems have been very popular. When the reason for this is sought, it will be found that hot water, as compared to steam or gravity-circulated warm air, offers a more uniform and steady heat. It is free from the unequal heat distribution and bulky duct work of the warm air system, and from the condensate troubles of the steam or vapor system. Like the steam system, it has no summer-time usefulness, and offers no possibility of air conditioning. The radiators and piping, furthermore, are larger than those for an equivalent steam system. In a hot water system the water is heated in a heater similar to a boiler except that it is completely filled with water, and no boiling takes place. Most hot water systems operate at about atmospheric pressure, which limits the temperature to which the water may be raised to slightly over 200° F. However, for heating homes, this temperature is entirely adequate for the coldest weather. Over much of the heating season, temperatures of 140 to 180° F are sufficient. From the heater the water passes to the radiators located in the rooms, where the heat is transferred to the air. Delivery of heat to the air cools the water which, as a result, becomes more dense than that in the supply mains, and it sinks to the heater; thus circulation is maintained by gravity and is due to the difference in density between the columns of hot and cool water between the radiator and the

heater. Central hot-water heating systems, and occasionally domestic systems, work on forced circulation created by a pump mounted at the heater inlet. This gives more flexibility to the system and permits the use of smaller pipes. The control of a hot water heating system is vested in a thermostat which operates the check damper and draft. The expansion elements of the thermostat are immersed in, or are subjected to the temperature of, the hot

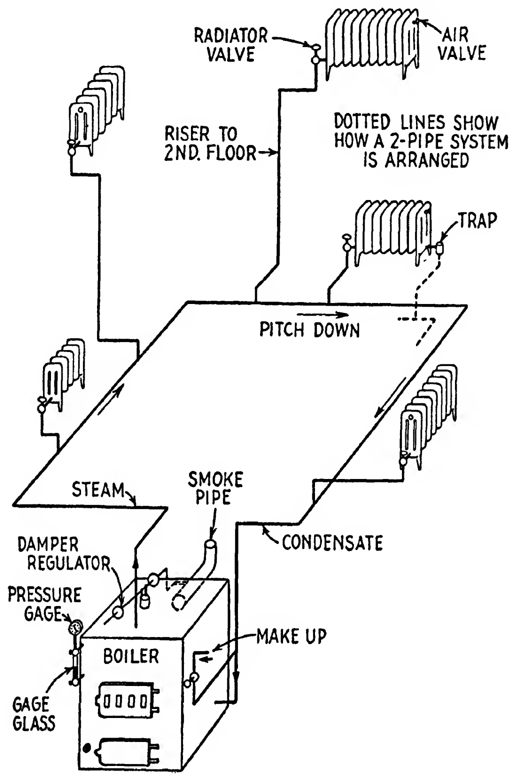


Fig. 7–15. Simple one-pipe steam heating system gravity condensate return.

water in the heater. (The thermostat tends to maintain uniform water temperature at the heater outlet by adjusting the rate of combustion.) To adjust the heat supplied to a room in accordance with weather changes, the water thermostat may be modulated from a wall thermostat located in a typical room or from an outside weather thermostat. In forced circulation systems the heater outlet temperature is maintained constant and a three-way valve is installed to by-pass some of the returning cool water into the supply main, thus tempering it in accordance with the needs for heat in the rooms. The room thermostat, in this case, would adjust the valve and so control the by-pass flow.

Steam heating, although lacking flexibility, and sometimes having difficulties of clearing condensate from the system, is, nevertheless, a convenient

medium for indirect heating because a considerable heating effect is obtained from a relatively small volume of the circulating fluid. Because the heat is released by condensing, the latent heat of evaporation is made available, whereas in hot water systems, only a portion of the heat of the liquid is available. Since latent heat is several times the amount of heat of the liquid, the

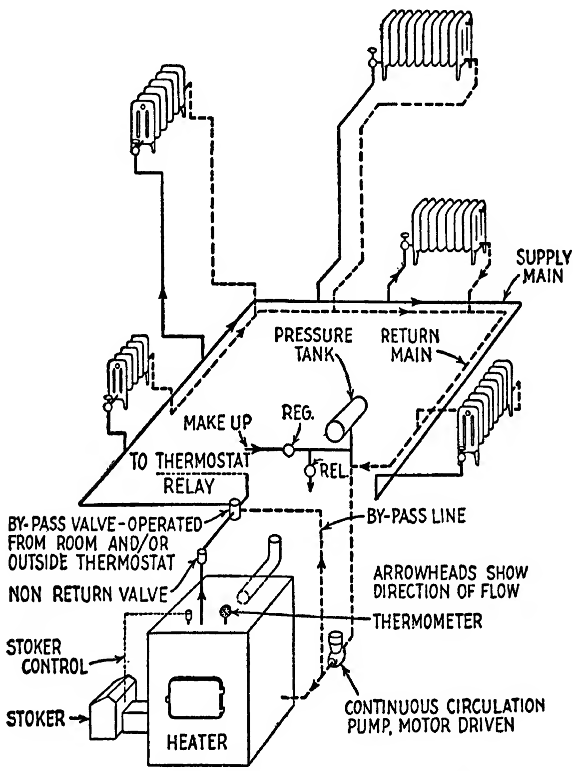


FIG. 7–16. Forced-circulation hot water system with thermostatically regulated heater by-pass (heater outlet temperature held constant).

size of a steam heating system compares favorably with any other of equivalent heating capacity. It is widely employed for large buildings, and to a more limited extent, for residences. As the circulation depends on the difference in density between steam and water instead of between hot water and cool water, it is quite positive. In laying out a steam heating system, it is very important to pitch the return pipes so that condensate will drain properly. In some cases the return and supply pipes are the same; that is, the condensate flows against the steam, but this is suitable only for small inexpensive installations. Two-pipe heating systems are found in many different arrangements involving differences of points of admission of steam to radiators, methods of expelling air, and returning of condensate to the boiler.

Although experimental at present, the "*heat pump cycle*" may possibly, in the future, receive favorable attention as a building heating system. The heat pump is a variant of the refrigeration process. In the normal refrigeration system the refrigerant is compressed, the resulting heat is removed by the condenser and then the refrigerant is allowed to expand to cool the stored material, i.e., it absorbs heat from this material. The heat pump uses this same principle, but utilizes a different part of the system for its useful function. It draws air or water from some outside source and cools it by having the refrigerant (if we may still call it this) absorb its heat. This refrigerant is then compressed and condensed. The heat removed at the condenser can be utilized for heating purposes. This system has the advantage that, by proper utilization of the apparatus, it may be used for summer-time cooling and the disadvantage of requiring a large motor for driving the compressor.

7–11. Ventilation. This is the process of changing the air in any space by natural or mechanical means. In its simplest form ventilation results from the effect upon buildings of winds, or from the natural levity of heated air. Air blown horizontally against a building creates a small plenum to windward and a small vacuum to leeward. Through openings intentionally created, such as opened windows, doors, ventilators, etc., this pressure difference will cause a ventilating current to flow. Also through cracks in walls, clearances around window sash, and the like, unintentional and unwanted ventilation may be brought into existence. Natural ventilation can be secured in the absence of atmospheric motion by providing elevated outlets (roof ventilators, stacks, air shafts) so that cool air may displace warm air upward and out of the vents.

Mechanical systems of ventilation are more amenable to continuous positive control of air flow. The vitalizing air pressure of these systems is usually created by a fan. Fans in this service are used to move air in predetermined directions for the purpose of changing the air in buildings, removing air charged with offensive odors, furnishing air to tunnels, etc. They are commonly fixed in position and connected to duct systems on the inlet or discharge sides, or both. Propeller fans have a limited use in this field, but ventilating fans are usually centrifugal because the application often requires overcoming of considerable static pressure.

TABLE 7–2. DUCT AIR VELOCITIES, FT. PER MIN. (AVERAGE PRACTICE)

Main ducts, dwellings, offices, etc.	1000–1200
Branch ducts, dwellings, offices, etc.	600–800
Grille and register openings, heating	200–400
Grille openings, air conditioning	500–1500 *
Main ducts, industrial	1500–2000

* Depends on "throw," or distance air is diffused from the outlet.

Ducts of sheet metal, wood, and other materials of construction carry the air to and/or from the space to be ventilated. Quietness in operation and even distribution of air (i.e., no drafts) are especially sought in these systems when installed in public buildings, dwellings, offices, and stores. Vibration absorbent mountings for fan and motor, low fan speeds, moderate air velocities, and acoustical duct linings are some of the means of reducing noise level. Air ducts are generally made with a rectangular cross-section if they are to be concealed in walls, ceilings, etc., since the available space is better fitted than would be possible with circular ducts. Cost and friction loss favor circular ducts, and they are used where concealment or head room space is not a determining factor.

Example: What size of ventilating duct would be needed to supply 4 air changes per hour to a department store floor covering an area of 10,000 sq. ft.? Ceiling height 12 ft. Space requirements limit the duct to a depth of 20 in.

Air flow per minute = $(10{,}000 \times 12) \times \frac{4}{60}$ = 8000 cu. ft. per min. Selecting a velocity of 1000 ft. per min. (Table 7-2), required cross-sectional area = $\frac{8000}{1000}$ = 8 sq. ft. Since the duct is limited to 20 in. one way, and a 20-in. circle does not provide 8 sq. ft., this will have to be a rectangular duct. Width of duct = $\frac{8 \times 144}{20}$ = 57.5 in., say 58 in. Duct dimensions: 20 in. $\times$ 58 in.

The minimum quantity of air in circulation consistent with human comfort has been the subject of extensive observations by the American Society of Heating and Ventilating Engineers and by others. In the usual space allotment of dwellings (approximately 400 cu. ft. per person) about $1\frac{1}{2}$ changes of air per hour will ventilate satisfactorily from both the chemical and thermal standpoint. This amounts to about 10 cu. ft. air per person per min. In workrooms, auditoriums, schoolrooms and other places of congregation, the free-space allotment per person is less and the required number of air changes greater.

TABLE 7-3. AVERAGE VENTILATING PRACTICE (FRESH AIR)

Ordinary room of a dwelling	1 air change per hr. and up.
Toilets, rest rooms, etc	2-5 air changes per hr.
Public dining rooms	4-10 air changes per hr.
Auditoriums	10-20 air changes per hr.
Stores	2-5 air changes per hr.

In conclusion, be it noted that ventilation is but one aspect of air conditioning, for the latter is concerned not only with quantity of air circulated, but also with its quality.

PROBLEMS

1. A heat pump used as a refrigeration compressor activates an air-conditioning system wherein 92,000 B.t.u. are extracted from the conditioned region per hour. The C.P. being 2.66, what horsepower is needed?

2. In an ice plant, 25 tons of ice were produced in 12 hrs. The compressor motor consumed 600 kw. hr. in the same interval of time. What was the overall coefficient of performance?

3. How many tons of refrigeration are represented by the situation shown in Figure 7–1? How much power is needed at the compressor? Compressor discharge pressure 180.6 psi. abs.

4. A brine tank is to be held at 20° F, with a temperature difference, brine to ammonia of 10° F. After compression the ammonia is condensed by water available at 60° F. Using tables, state what the low and high pressures of the cycle are. Record any assumptions made.

5. In a domestic refrigerator using Freon, what condenser temperature difference is available on a day when the room temperature is 80° and the compression pressure is 100 psi. gage?

6. Solve the examples associated with Figure 7–1 for methyl chloride as the refrigerant.

7. Repeat Problem 6, but with Freon as the refrigerant.

8. Diagram a refrigeration cycle such as Figure 7–2, but for an ice-making plant. Draw both the flow diagram and the T–s diagram (no scale). The refrigerant is ammonia operating between 48 and 160 psi. abs. At significant points on the flow diagram indicate the ammonia temperature and physical state.

9. If the ammonia vapor of Problem 8 leaves the coils dry and saturated, what are the temperature and heat content after an isentropic compression?

10. Methyl chloride has a quality of 10% after passing through an expansion valve into refrigerating coils at 30 psi. abs. After evaporation, the dry vapor is compressed to 80 psi. abs., 130° F. Calculate the coefficient of performance.

11. Compute T and s for all control points of the cycle of the domestic refrigerator described on page 158, for ideal processes. Plot the cycle to the scale of 1 in. = .04 entropy, 1 in. = 20° F. Break the T axis at 0° F.

12. What is the C.P. for the data of Problem 11?

13. Plot a T–s cycle for ideal ammonia vapor refrigeration. Evaporating temperature 40° F, condensing temperature 80° F. 1 in. = .3 entropy, 1 in. = 20° F. Break the T axis at 0° F.

14. What are the temperature and volume of one pound of H_2O, originally at 50° F and 14.7 psi. pressure, after being flashed to an absolute pressure of 0.23 in. Hg?

15. Repeat Problem 14 for a chamber at 29.54 in. Hg vacuum, barometer 29.78 in.

16. Using the steam tables, determine the foot-pounds of work necessary to compress a pound of steam isentropically from a saturated condition at 42° F to 0.5 in. Hg. Compare this with the foot-pounds required to raise it through the same pressure range as a liquid.

17. Air having an absolute humidity of 140 grains per lb. is cooled to 60°. Would any moisture have condensed from it? If so, how much?

18. Compute the pounds of moisture per pound of dry air for a saturated atmosphere at 50° F.

19. Using Avogadro's principle, show that if air is a mixture of 23.2% O_2 and 76.8% N_2 by weight, it becomes 20.9% O_2 and 79.1% N_2 by volume.

20. You know that a cooling tower was able to cool water from 120° F to 90° F at the rate of 1200 gals. input per hr. Estimate the gallons of make-up water this tower would have to receive per 24-hr. day when cooling at this rate.

21. A cooling tower is used to cool the condensing water of an ice plant. Water is heated from 60° to 75° F in the condenser. Diagram the tower, label with all known temperatures, estimate the percentage of make-up.

22. Consider a two-story building to be one region for the purpose of computing air-conditioning cooling load. It is cubical, 50 ft. × 30 ft. × 20 ft. high, with a flat roof. Glass area 20% of wall area. Occupancy 400 sq. ft. floor area per person. Walls 8-in. cinderblock, ground floor concrete, roof of insulated concrete slab with gravel and asphalt surface. Maximum outside temperature 100° F, inside temperature 10° less. Ventilation, one air change per hour. Find tons of refrigeration required.

23. Calculate the B.t.u. per hour cooling load for conditioning your classroom, assuming remainder of building also conditioned, and all seats occupied. Ventilation, 5 cu. ft. per min. per person. Differential 15° F.

24. The sensible heat to be taken from a classroom is 25,000 B.t.u. per hr. With a diffusion temperature of 10° allowable, how much conditioned air must be supplied? Answer in cubic feet per minute. If 25% of the supply is fresh, how many occupants would this provide for at the rate of 10 cu. ft. per min. each?

25. Using the pertinent data of Problem 22, calculate the square foot EDR for a steam system to heat that building to 70° F on a zero degree day.

26. How much surface of hot-water radiation is needed for heating the classroom to 70° F on a 10° day? Allow four air changes per hr., and credit 50% occupancy at the rate of 200 B.t.u. per hr. per person.

27. If the classroom is radiator heated, obtain data on the square foot EDR per section and determine the hourly heat liberation in the room. Make a study to determine the adequacy of the installed radiation, i.e., how many air changes will it care for and how do these compare with data of Table 7–2?

28. Diagram a two-pipe steam heating system. Single story building, boiler in basement.

29. Diagram a ventilating duct to convey fresh air into the classroom, using 10 cu. ft. per min. per occupant, or 10 air changes per hr., whichever is least. Duct velocity 700 ft. per min., depth 12 in., grille velocity 300 ft. per min.

30. What diameter is required in a circular ventilating duct supplying an auditorium seating 1200 persons?

CHAPTER 8

Spark Ignition Engines

8–1. Internal Combustion Engine. The spark ignition engine is one form of the internal combustion engine. The compression ignition engine, treated in Chapter 9, is the other form. The internal combustion engine is one in which combustion of fuel takes place within the cylinder of a displacement machine, and the products of combustion form the working medium during the power stroke. As it is a self-contained power supply unit, the internal combustion engine assumes a position of great importance in the power field, especially for that class of service where portability, light weight, and compactness are important. Witness the majority of self-propelled vehicles powered by internal combustion engines. The principal cycles of internal combustion engines in use at present are the *Otto* and *Diesel* cycles. Other cycles are of some historical importance, but these two have supplanted the others in the course of time by possessing certain points of superiority such as economy, reliability, compactness, etc.

An internal combustion engine consists primarily of a cylinder and a piston, generally single acting, which, together, form a combustion chamber of variable volume. Both of these parts are constructed of metal, and, as the temperatures attained during combustion are well above the ability of uncooled metals to withstand, the cylinder of an internal combustion engine must be adequately cooled by transferring through the cylinder wall a certain amount of the heat contained in the gases of combustion. This is accomplished in a practical way by surrounding the cylinder with a jacket of cooling water, or by providing it with an extended outer surface of fins so that air can absorb enough heat to keep the metal cool. The required motion is given to the piston by a crank and connecting rod mechanism which also serves to take from the piston the power developed by gas pressure.

An engine operating with flaming gas in its cylinder would not last long with simple metal to metal contact of the moving parts, therefore a lubricating system, embodying oil as the lubricant, is an important feature of every internal combustion engine. The most difficult job is that of lubricating the piston in the cylinder. During a portion of the stroke, at least, the lubricated wall is exposed to incandescent gases which tend to burn off the film of lubricating oil. The cooling system must maintain the metal surfaces cool enough to save that film.

The events of the cycle upon which an internal combustion engine works are controlled chiefly by the operation of valves located in ports leading to and from the cylinder. Generally, an admission or inlet valve, and an exhaust valve, are provided in each cylinder. The operation of these valves is derived mechanically from the crankshaft through the valve gear system.

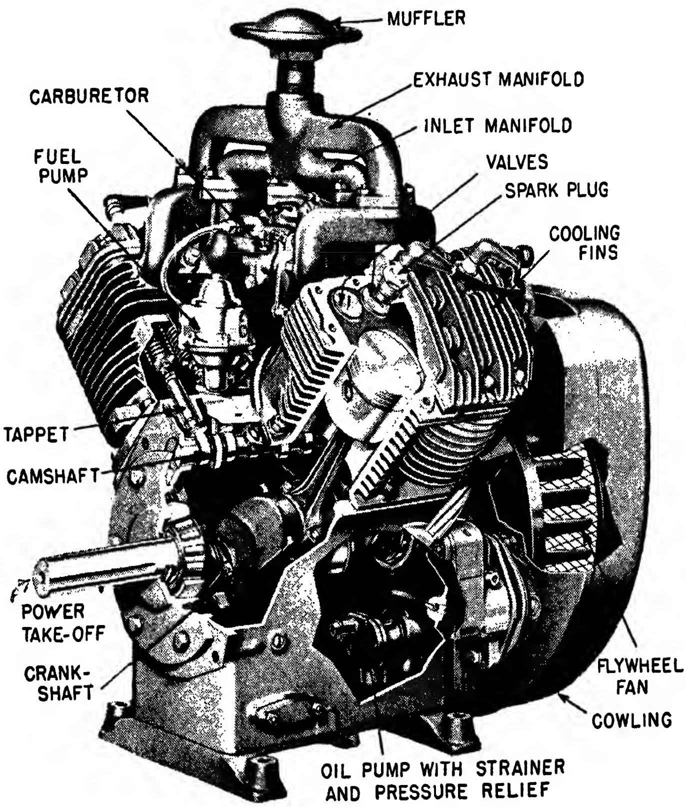


FIG. 8-1. Cutaway view—spark ignition engine. Four cylinder vee type, four-cycle, air-cooled. (Courtesy Wisconsin Motor Corp.)

The combustion of fuel in an internal combustion engine is not a continuous action, but a series of individual explosions, each one requiring a metered amount of fuel to be individually ignited. For this reason, every internal combustion engine must incorporate an ignition system, whose function it is to supply in proper time the ignition temperature required for combustion. The internal combustion engine is of a type tending to deliver its power cyclically, and in a fashion which would be very fluctuating unless balanced by the use of a heavy flywheel, or by overlapping of power impulses through multi-cylindered arrangements. In fact, it is usual to build internal combustion engines with more than one cylinder so that the delivery of power will

be more uniform, and flywheel proportions will not be excessive. The supply of fuel to multi-cylindered engines from a common source, and the conduction of exhaust from them, leads to another service feature for the internal combustion engine—namely, the inlet and exhaust manifolds.

The production of power by this type of engine represents a thermodynamic conversion into mechanical energy of a portion of the heat energy developed. Heat energy enters the engine potentially in the form of fuel. Mechanical energy appears as power available at the crankshaft. Unavailable or rejected heat is found in exhaust, cooling, and friction. The conversion of the energy of the fuel into useful power takes place about as follows: Air is brought into the cylinder and, either after, before, or during compression, depending on the cycle, fuel is introduced into the air and mixed with it. Upon ignition of this fuel, the heat developed raises the pressure of the products of combustion, or, at least, maintains the pressure during some motion of the piston. The piston has against its face a high gas pressure, which results in the transmission of energy through the train of mechanism consisting of moving piston, wrist pin, connecting rod, and crankshaft. During the motion of the piston, the gases of combustion expand and are cooled somewhat. It has not been found economical to build an engine sufficiently bulky to expand the gases until they reach ordinary atmospheric temperature, and there is always considerable heat lost in the exhaust. In spite of the losses the internal combustion engine of the present is a practical prime mover.

Summarizing the above, it appears that the internal combustion engine can be functionally subdivided thus:

1. The piston and cylinder, forming an expansible *combustion chamber*.
2. A *mechanical linkage* to transfer the gas pressure against the piston into a rotating torque. This consists of piston, connecting rod, and crankshaft.
3. Valves and *valve gear* to produce the combustion cycle.
4. A supply of fuel and air, properly conditioned. The *carburetion system* supplies this service for spark ignition engines.
5. A means of timed ignition—an *ignition system.*
6. A *lubrication system.*
7. A *cooling system* to protect the lubrication system and prevent overheating of the combustion chamber walls.
8. A *speed control,* and regulating system. Governors, heavy flywheels, multiple cylinders.
9. Some means for initiating the cycle of events which when started is self-perpetuating, i.e., a *starting system.*
10. A base, *frame,* crankcase, or bedplate which is a foundation for mounting the other components in the proper relative positions.

8–2. Otto Cycle. The possibilities of an internal combustion engine which would operate on the four-stroke cycle were analyzed in the nineteenth century by de Rochas. The action of the engine, which was at that time described only, is in all fundamental respects the same as that used today in spark ignition engines. Shortly after this internal combustion engine analysis was made, the German Otto built an engine (1876) which operated on a four-stroke cycle, and which has come, since, to be called the Otto cycle. This cycle is carried out by a series of operations, namely, and in order:

1. *Suction* during outward stroke of the piston.
2. *Compression* during inward stroke of the piston.
3. *Power* during outward stroke of the piston following ignition at inward dead center.
4. *Exhaust* during inward stroke of the piston.

Although the first engines built by the Otto works were four cycle, the cycle can be adapted to two-stroke cycle action. An engine which operates on the Otto cycle has considerable clearance volume into which the air and fuel drawn in on the suction stroke are compressed. As the piston pauses instantaneously on dead center position, the charge is ignited and combustion of the charge occurs at constant volume. The heat added by this combustion greatly increases the pressure. During the expansion stroke, this pressure is lowered, and a considerable amount of the heat of the gas is converted into work. At the end of expansion, the pressure is dropped to the exhaust point, and the exhaust gases are pushed out of the cylinder, which is thus cleared for the next suction stroke. In the figure which illustrates the Otto cycle in theory and in practice, the cycle is shown on the pressure-volume plane. The ideal cycle is shown solid, and departures from it, made by the actual cycle, are shown dotted. The ideal four-stroke cycle is made up in part of two isometric changes, one representing the combustion of fuel at constant volume, the other the rejection of heat to the exhaust at constant volume. There are two adiabatic changes, one representing the compression of the charge, the other the power stroke. Finally, there are two coincident constant pressure

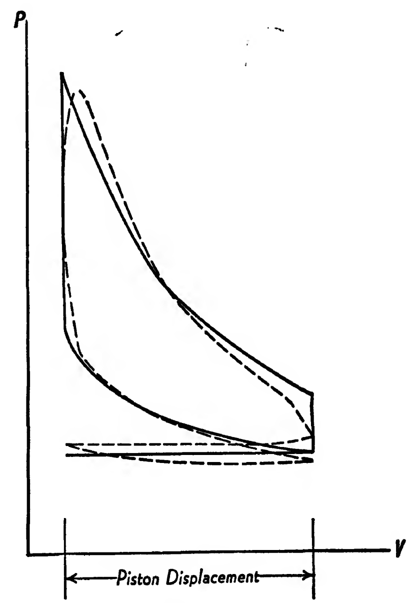


FIG. 8–2. Otto cycle diagram.

changes, representing the ideal exhaust and suction strokes. An actual working cycle will not have quite the same processes. Combustion of a fuel is not instantaneous in nature. That occurring in the gasoline engine occupies an appreciable length of time compared to piston motion, so that isometric pressure rise is not possible. The point of ignition must be advanced before inward dead center, or there will be considerable loss of work by delayed combustion. An exchange of heat between cylinder cooling water jackets and cylinder contents tends to destroy the adiabatic relationship assumed for the ideal cycle. The exhaust cannot be at constant pressure because of the inertia of gases leaving the cylinder and the definite rate of valve action of mechanically driven valves. Furthermore, the exhaust and suction lines cannot be coincident, since exhaust must be above atmospheric pressure by the amount of friction losses through the ports and manifolds, while suction must be below atmospheric pressure for the same reason.

8–3. Air Standard Efficiency. The actual thermal efficiency of internal combustion engines depends on many indeterminate factors which render the rational computation difficult, if not impossible. A hypothetical efficiency may be computed, based on certain assumptions, as follows: *first*, that the internal combustion engine has no mechanical friction; *second*, that the compression and expansion in the cylinder are those of pure air (whereas actually the expanding gases are composed of nitrogen, oxygen, steam, carbon dioxide, and carbon monoxide, and the gas compressed is never pure air); *third*, that the compression and expansion are adiabatic. This would imply a heat insulation jacket around the cylinder, but the difficulties of successful lubrication have required all practical internal combustion engines to be positively cooled. Based on these three assumptions, thermodynamic theory can be used to yield an equation of efficiency of the cycle, and such is termed the "air standard efficiency." The actual thermal efficiency will, of course, be considerably less than the air standard efficiency. Nevertheless, the air standard efficiency is useful as a measuring stick for the various designs, and it also shows the effect, on efficiency, of varying the compression ratio. The air standard efficiency of the Otto, or gasoline engine cycle, is given by the equation:

$$\eta = 1 - \frac{1}{r^{\gamma-1}}.$$

r is the compression ratio,* and γ the ratio of the specific heats at constant pressure and constant volume. The actual efficiency of the Otto cycle is usually between 20% and 25%. The compression ratios employed have been steadily raised by manufacturers in order to increase the efficiency of their product, for, as is seen in the above equation, an increase of r will increase the efficiency of the Otto cycle engine.

* Chapter 5.

Continuous improvement of efficiency by the use of ever higher compression ratios is limited by the disagreeable tendency of the fuel to detonate at the higher compression ratios. Modern gasoline engines will be found with compression ratios ranging from 5.5 to 7.0.

Example: What is the air standard efficiency of an Otto cycle operated with a clearance of 25%?

The compression ratio is fixed by the clearance. In this case

$$r = \frac{1 + .25}{.25} = 5. \quad \gamma - 1 = 1.4 - 1 = 0.4.$$

$$\eta = 1 - \frac{1}{5^{.4}} = 47.5\%.$$

8–4. Mechanical Action. A typical multi-cylindered engine is arranged with cylinders in line, or in banks, or radially. The cylinders are aligned so that the connecting rods may bear on the same crankshaft. This crankshaft drives the necessary auxiliaries, and delivers the remaining power as useful output at a coupling, pulley, or clutch. Each cylinder has at least one inlet and one exhaust valve mechanically operated and synchronized with the cycle. The ignition apparatus is located in the combustion chamber either as an igniter head or a spark plug. The openings to the cylinder, which are opened and closed by the valves, lead to ports, to which manifolds are connected. Auxiliaries which must be driven by the engine, and which consume part of its gross output, are the aforementioned valves, the lubricating oil pumps, cooling water circulating pumps, the generator for current to the ignition system, the igniters or timers of the ignition system, and sometimes an air fan. The cylinders are mounted on a crankcase, which provides alignment, and also supports the crankshaft bearings.

The operation of an Otto engine may be described beginning with the suction stroke. At the beginning of the suction stroke, the inlet valve will be open, and will so remain until the outward stroke is completed, drawing in through the carburetor and manifolds a mixture of gasoline and air in explosive proportions. The quantity drawn in per stroke depends on the efficiency with which the cylinder is charged by the induction system. On the succeeding inward stroke of the piston the inlet valve remains open for possibly 10 to 15 degrees of crankshaft travel. There is a certain inertia of the gas column moving in the manifolds and ports, so the flow will not immediately reverse upon reversal of the piston travel. The inlet ports should remain open as long as any gas will flow into the cylinder, so that the charging efficiency will be maximum. After closure of the inlet valve, the gas is compressed into the clearance space at the extremity of the cylinder. This clearance space is the combustion space, and its shape has been given a great deal of study in con-

nection with detonation. In this space is located the spark plug which receives the high-voltage impulse from the ignition system when the crank pin is still 10 to 15 degrees (sometimes even more) before dead center. The advance of this spark before dead center is variable, so that it may be advanced more at high rotative speeds, thus aiding completion of the explosion before expansion begins. On the outward stroke, which is the power stroke, both valves remain closed until the crank is about 30 degrees before outward dead center, when the exhaust valve begins to open. The valve is given this lead on the piston position in order that it may be fully open at the dead center

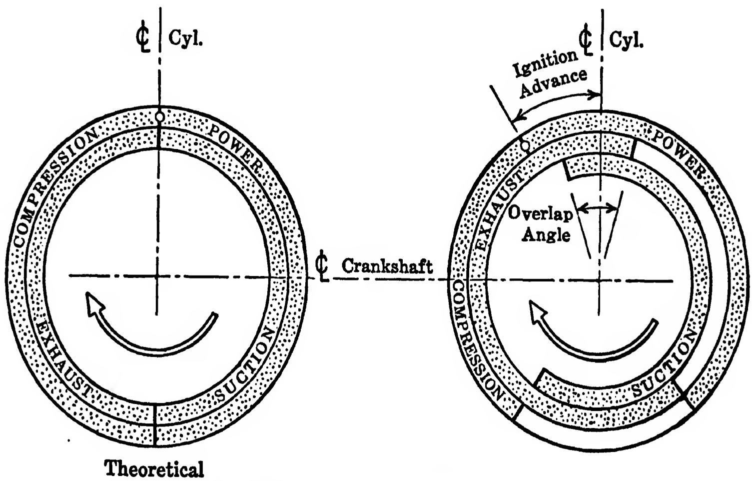


FIG. 8–3. Valve-timing diagram based on crankpin motion (four-cycle engine).

position, so that on the inward exhaust stroke a complete scavenging of burned gases from the cylinder can be a possibility. In fact, the exhaust valve remains open until after the piston is slightly past inward dead center position.

The "events" of the engine cycle, such as beginning of exhaust, end of suction, etc., are, of course, under the control of the mechanism which actuates the valves. Both valves and valve gear suitable for spark ignition engines were described in Chapter 5. It is nearly universal practice to operate poppet valves by cams. In the usual multi-cylinder design several, sometimes all, cams are machined on one camshaft. The builder must necessarily have adopted some cycle timing diagram, such as Figure 8–3, and constructed the cams on the shaft accordingly so that each cylinder was identically valved, although of course not in unison. An examination of Figure 8–4 will show the positions involved in meshing the camshaft gear with the crankshaft gear. Suppose the timing is related to the event of beginning of exhaust on cylinder No. 1, the same to occur at a crank angle of 40° before lower dead center (angle θ). The crankshaft can be placed in position for piston at top dead center by removing the cylinder head and revolving the flywheel until it is

seen that the piston is at its upper dead center position. Rotation through 140° then brings the crankshaft to the position it should occupy when the exhaust valve begins to open. It is necessary that the cam have lifted the tappet a distance equal to the valve stem clearance before the valve is ready to open. When the camshaft has been rotated to this position the gears are correctly meshed. Small timing marks may be stamped on the gears to indicate the proper position. These are shown in Figure 8-4, however manufacturers usually mark the teeth that will be in mesh when the piston of cylinder No. 1 is at TDC beginning the power stroke.

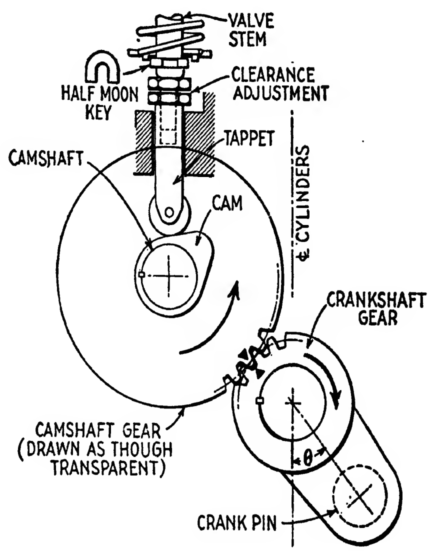


Fig. 8-4. Valve-timing marks.

Any discussion of valve operating gear would necessarily be based on some knowledge of common cylinder head arrangements. These might be classified as T, L, I, and F arrangements, these letters being symbolic of the cross-sectional shape of the cylinder head. In the T head, the combustion chamber has pockets on either side, in one of which is located the exhaust valve, in the other the inlet valve. Earlier engines were so built, but the construction has been abandoned because two camshafts were necessary. The L-head arrangement is the most common today. The valves are both located in the same pocket on the same side of the cylinder. In-line multicylinder engines, then, have all the valves in a single line, so that they may be driven by cams, all of which are machined on a common camshaft. When the valves are located overhead in the top of the cylinder instead of in a special valve pocket, the arrangement is known as the I head. The valves open downward into the cylinder in the I head, whereas in the T and L they open upward. I-head arrangement has advantages of more direct gas flow to the cylinder, thus promoting higher volumetric efficiency; but has the disadvantage of a more complicated valve gear. In the F head, which has

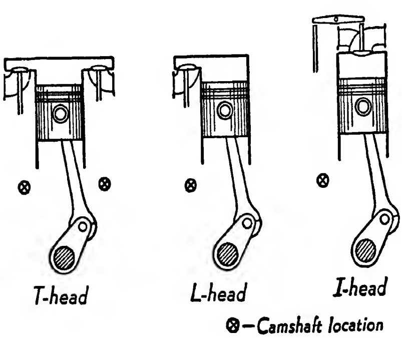


Fig. 8-5. Common cylinder shapes as governed by valve position.

been used but little, the exhaust and inlet valves are mounted one over the other in a valve pocket, one being similar to an L-head valve, the other to an I-head valve.

Valves in L-head engines are set directly in line with the cam shaft, so that the cam either bears directly on the end of the valve stem, or indirectly, through a relatively short follower. As the cam provides positive motion in one direction only, the valve is opened against a spring pressure. The valve spring holds the cam follower against the cam. On I-head engines, the cam shaft must either be placed above the cylinder, a somewhat awkward position, or motion must be transmitted from its crankcase location to the valve by push rod and rocker arm.

The foregoing remarks apply, in the main, to the four-stroke cycle engine. In some respects they apply to the two-stroke, but there are essential points

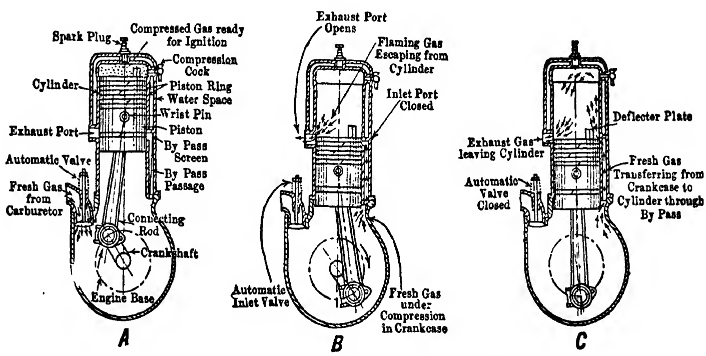


FIG. 8-6. Two port two-cycle engine.

of difference. The two-cycle engine is characterized by the absence of valve gear, since the piston moves to uncover ports in such a way that the events of admission and exhaust are accomplished without the use of valves. Figure 8-6A represents the two-cycle engine nearing the inward dead center, with the compressed gas ready for ignition. Ignition occurs, and the resulting combustion raises the pressure and drives the piston outward on a power stroke. Near the end of this outward stroke the piston uncovers an exhaust port in the lower part of the cylinder wall, and the residual pressure drives the exhaust gas quickly out through this port. Further outward motion to the dead center position then uncovers a port leading from the crankcase to the cylinder. The crankcase is filled with a slightly compressed mixture of gasoline and air, which is rapidly passed through this port. Entering the cylinder, being deflected upward, it tends to drive the remaining burned gas out through the exhaust port, which is still open. As the piston moves on its inward stroke, it closes these ports and compresses the gas into the clearance space. The same motion tends to create a vacuum in the air-tight crankcase, with the

result that a fresh charge is introduced into the crankcase to be compressed on the down stroke. The comparatively shorter length of time available in this engine for charging and scavenging the cylinder makes it impossible to attain volumetric efficiencies * comparable with the four-stroke cycle, except for relatively small, slow-moving engines.

8–5. Combustion. When examining the numerous mechanical details of the S.I. engine let us not forget that the engine was built for one purpose: the conversion of heat into work. Production of that heat is therefore a matter of primary importance. Spark ignition engines may be built for any fuel

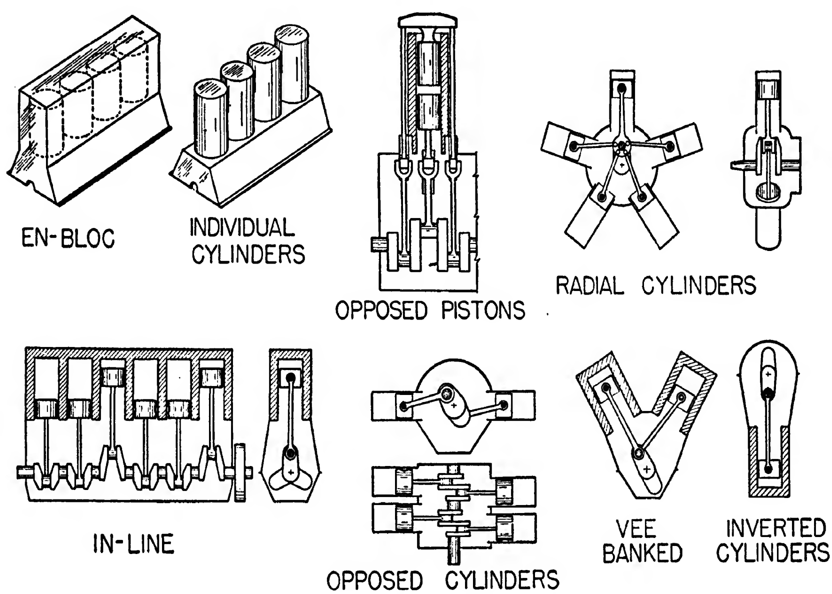


FIG. 8–7. Piston and cylinder arrangements.

which will burn explosively and which can be thoroughly mixed with air without too much difficulty. Gaseous fuels, alcohol, and several of the highly volatile hydrocarbon mixtures in petroleum may be used. Based on current usage, gasoline is, by long odds, the principal fuel for this type of engine. Consequently, "gasoline engine" and "spark ignition engine" are, from a practical viewpoint, equivalent descriptions.

Gasoline † being a liquid fuel must be vaporized before a satisfactory mixture with air can be achieved. Being highly volatile, the spraying of it into an air stream results in a satisfactory mixture. The vaporization and metering of a liquid fuel so as to produce an explosive mixture with air is known as *carburetion,* a subject we will examine at some length in a subsequent section.

* This term is defined on page 223.

† See Chapter 3 for information on the nature and production of gasoline.

Assume now that an explosive mixture is made and inducted into the cylinder on the suction stroke. The following compression heats the mixture somewhat but not enough for spontaneous ignition, otherwise ignition would be uncontrolled and irregular. As the piston nears the end of its compression stroke an electrical potential of several thousand volts is "planted" on the spark plug electrodes by the ignition system. The high voltage breaks across the gap of a few thousandths of an inch creating a short but intensely hot *spark* which raises some hydrocarbon and oxygen molecules to the temperature where they can begin to combine in combustion. Once started in an explosive mixture, combustion is self-propagating at extremely high speeds, so all the mixture is quickly burned.

The combustion of gasoline in the Otto engine is as follows: (Gasoline is a mixture of many hydrocarbons, for which the formula C_8H_{18} is taken as a fair average composition.)

$$C_8H_{18} + 12.5\ O_2 = 8CO_2 + 9H_2O.$$

Calculations based on this equation show that 15.2 lbs. of air are required for the complete combustion of each pound of gasoline. A deficiency of air results in the formation of carbon monoxide and hydrogen, but if there is only a small deficiency, the hydrogen may be assumed as being completely burned. Theoretically, oxygen and carbon monoxide should not be present in the combustion of a perfect mixture, but actually oxygen can be found in the combustion of rich mixtures, and carbon monoxide will persist in the products of combustion of lean mixtures. The limits of explosive mixtures of gasoline and air are about 9 and 20 lbs. of air per pound of gasoline. Maximum power is derived at about 13 lbs. of air, the ideal conditions call for 15.2 lbs., and maximum economy is obtained with even leaner mixtures.

The higher heating value of gasoline is stated in Chapter 3 to be 20,000 B.t.u. per lb. The sensible heat, which is lower heating value, is about 19,500 B.t.u. This is the heat from which the work is obtained because the exhaust products are always sufficiently hot so that the latent heat remains in the water vapor. Although most of the 19,500 B.t.u. will ultimately be developed by combustion, it is not to be expected that this will happen immediately after ignition. Gasoline engines use mixtures of nearly ideal proportions and the temperature rises quickly until dissociation of the products is in equilibrium with combustion. As the piston travels on its power stroke the expansion cools the products and allows the dissociated fraction to burn. Since this residual combustion occurs after the isometric phase is completed, it (1) transfers heat to the working medium during what is ideally an adiabatic process and (2) its heat is not as "available" for conversion into work as the isometric heat. The complete analysis of S.I. engine combustion is one of the more difficult problems of applied thermodynamics, as it must consider (1) dilution

of the incoming charge by burned gas occupying the clearance space at the end of the exhaust stroke, (2) variable specific heats, (3) dissociation, and (4) the quality of the mixture, i.e., lean, normal, or rich.

Combustion chambers for these engines are comparatively simple. Details vary as to location of spark plugs and position of valves. As compression ratios were increased, prevention of detonation became a major problem. Much research has been carried out to determine the manner in which detonation may be a function of combustion chamber shape. This has resulted in various arrangements of spark plug location, as well as special shaping of the interior curves of the cylinder-head surface in order to promote turbulence, reduce length of flame travel from the spark, and secure good heat transmission from gas to cylinder head.

Example 1: After completion of the suction stroke the mixture in a cylinder has a pressure of 14.0 psi. abs., and a temperature of 80° F. What potential heat value has been inducted per cubic foot of displacement?

Density of air at the stated conditions is determined by the general gas law,

$$\frac{w}{V} = \frac{14 \times 144}{53.4 \times 540^\circ} = .0698 \text{ lb. per cu. ft.}$$

If the air-fuel ratio is 15:1, a good approximation of the weight of fuel per cubic foot of mixture is $.0698 \times \frac{1}{15} = .00465$ lb. The lower heating value of gasoline being 19,500 B.t.u. per lb., the potential heating value of a cubic foot of the mixture is $.00465 \times 19{,}500 = 90.6$ B.t.u.

Example 2: What is the ideal air-fuel ratio for an engine burning methane, CH_4? What is the heating value of a cubic foot of the mixture under standard conditions?

In Table 3–2 the air-fuel ratio is given as 17.2:1. Mols of mixture present with one pound of fuel are $(17.2/28.8) + \frac{1}{16} = 0.658$ mols. Volume of that quantity $= 380 \times .658 = 250$ cu. ft. under standard conditions. The same table of data gives lower heating value 21,375 B.t.u. per lb. Since 250 cu. ft. possess 21,375 B.t.u. heating value, the B.t.u. per cubic foot are 85.3.

8–6. Detonation. Under certain conditions a metallic knock, or ping, may be heard in spark ignition engines. The technical name for this action is detonation. Detonation is undesirable. It reduces power output, causes overheating, unduly stresses the cylinder head, and is generally objectionable from the noise and vibration standpoint. It decreases thermal efficiency.

Detonation has been extensively studied. Apparently it is due to spontaneous ignition of the explosive gasoline-air mixture in the combustion chamber. Observers note that with no detonation, ignition of the charge starts at the spark plug and travels rapidly, but with a definite wave front, through the charge. However, when the engine is knocking, the inflammation proceeds normally only part way from the source of ignition, and then suddenly the remainder of the charge is ignited simultaneously at all points, accompanied by a sharp rise in pressure due to almost instantaneous expansion. The

result is a heavy pressure wave striking the cylinder head a hammer-like blow. The explanation advanced for this sudden ignition of the entire charge is that the initial combustion increases the pressure of the rest of the charge, and its temperature, until it passes the point where ignition will take place spontaneously due to adiabatic compression.

Conditions affecting detonation might be discussed under three heads; viz., fuel characteristics, cycle characteristics, and engine characteristics. Since the ratio of air to fuel in the combustible mixture also affects the rate of burning, it has a bearing on detonation. The spontaneous ignition temperature of the fuel is another important characteristic. Turning next to the cycle upon which the engine is operating, we find the compression ratio and the time of ignition to be of great importance. The overcoming of detonation is of increasing importance, as compression ratios of engines are raised in the effort to increase thermal efficiency. Detonation may be offset by retarding the spark timing. But this is undesirable, since it is accompanied by loss of power and overheating. One of the more important engine characteristics affecting detonation is the material of the combustion chamber. The more rapidly heat is conveyed away from the cylinder head, the less will be the chance of the initial combustion raising the pressure above that for spontaneous ignition. Thus aluminum cylinder heads and improved cooling efficiency offset detonation in high compression engines. Manufacturers' provisions for thus conducting heat away rapidly will be nullified by a heavy layer of heat-insulating carbon deposit in the cylinder head. Other engine characteristics of importance are the location of the spark plug and the shape of the combustion chamber.

After extensive experimentation it was found that certain substances had the ability to suppress detonation. How they act is not fully known. Amounts of suppresser as small as one molecule to 100,000 molecules of explosive mixture are effective in eliminating detonation.

Lead tetraethyl in small quantities is effective as a knock suppresser. To prevent a lead deposit from forming in the cylinder head, ethylene dibromide is added to form a lead bromide which is powdery and is blown out through the exhaust ports. Gasoline so treated is called ethyl gasoline.

The detonating quality of a gasoline is one of its comparable characteristics. A scale called octane rating has been devised to measure this quality. The *octane rating* is the percentage by volume of isooctane in a heptane-isooctane mixture, which exactly matches in anti-detonating property, an actual fuel under test in a standard test engine at standard conditions.

As detonation limits the compression ratio that may safely be allowed in engine design, so also it limits the possible efficiency of the Otto cycle. This is a clue to the interest taken in it by manufacturers, research engineers, and all others seeking to improve this type of engine.

8–7. Carburetion. The vaporizing and mixing of a liquid fuel with air in the correct proportions is called carburetion, and the device to accomplish this, a carburetor. The automobile engine offers an instance of applied carburetion, also one of the more difficult carburetive problems, since it is expected to operate smoothly and economically with wide variation of power, using fuels of varying quality.

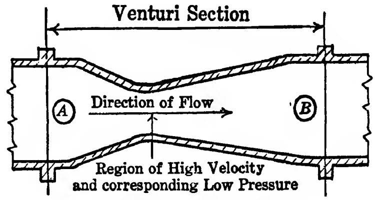


FIG. 8–8. Venturi tube.

The basic spray and metering action of most commercial carburetors is obtained by *Venturi* action. An incompressible fluid flowing through a tube having a throat must necessarily speed up some to get through the small section. From Bernoulli's Law (Chapter 4) we know that, in a case like this, where velocity is increased, pressure is consumed. The Venturi tube is simply a section containing a throat with a smoothly rounded entrance followed by an efficient diffuser so that the original pressure will be regained. Since the static pressure at the throat is a function of the rate of flow, it can be used for metering purposes. Air is expanded so slightly by Venturi tube pressures that it may be considered to meet the incompressibility requirement.

In carburetion, the Venturi is located in the passage leading from the atmosphere to the cylinder. As the air passes through the throat its static

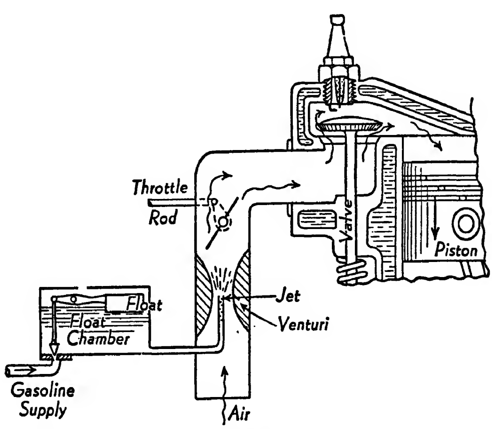


FIG. 8–9. Induction system of the gasoline engine showing elementary form of carburetor.

pressure is reduced. A gasoline chamber, with a gasoline level which is maintained slightly below that of the level of the throat of the Venturi, supplies gasoline through an interconnecting tube to a jet whose opening is located in the area of reduced pressure at the Venturi throat. Since the surface of the

gasoline in the float chamber is subject to atmospheric pressure the pressure difference causing gasoline to be sprayed out of the jet is the same as that which exists between the atmosphere and the throat of the Venturi tube. Thus both air and gasoline are subjected to the same driving pressure, and when one increases the other increases, and vice versa. If the discharge of gasoline from the jet, and of air through the Venturi increased in the same ratio, this simple carburetor would be successful for automotive service. Unfortunately the rate of increase of gasoline flow with increasing pressure difference exceeds that for air.

The *mixture* delivered by this carburetor is given by the following equation, in pounds of air per pound of gasoline:

$$\text{Air-fuel ratio} = \frac{C_a A_a}{C_g A_g} \sqrt{\frac{d_a}{d_g}},$$

in which C = Coefficient of discharge.
A = Cross-sectional area of flow at the Venturi.
d = Density of the fluid.

With ordinary gasoline this mixture should be maintained at a theoretical 15.2 lbs. of air per pound of gasoline. The simple Venturi tube carburetor will not do this, but rather will give an increasingly richer mixture, as the velocity of air through the Venturi increases. The principal reason for this characteristic will be found in the variability of the coefficients of discharge. The commercial Venturi-type carburetor contains modifications which offset this basic undesirable characteristic so that a suitable mixture will be maintained at all loads.

Although a mixture of 15.2 lbs. of air per pound of gasoline is theoretically correct, mixtures as rich as 9:1 or as lean as 20:1 are explosive. However,

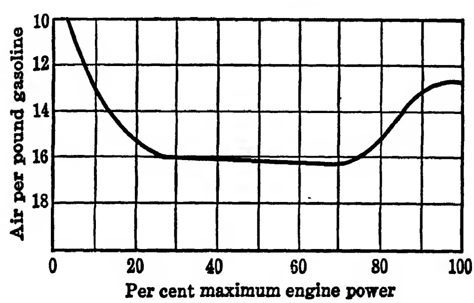


FIG. 8–10. Desirable carburetor performance.

the rich mixtures are uneconomical, and the lean mixtures must be employed cautiously at maximum power to prevent detonation and overheating. Thus rich mixtures are indicated for full power operation. Also, on account of the

proportionately high dilution of incoming fuel stream by unscavenged products of combustion of the previous cycle at light loads (under quantity control), a rich mixture is needed at low power output. Between these needs for rich mixture, the carburetor should deliver a fairly uniform lean mixture for the sake of fuel economy. The graph on page 198 illustrates a desirable performance for the carburetor of a gasoline engine. By employing multiple jets, adjustable orifices and other intricacies, commercial carburetors attain mixture control approximating this desired performance, but at considerable sacrifice of the simplicity of the principle of Venturi tube metering.

8-8. Carburetor Types. A carburetor is any device for securing the carburetion function just described. It is a necessary device for any engine, furnace, kiln, etc., where a volatile liquid fuel is prepared for combustion by vaporizing it into the flow of combustion air. The possibility of accomplishing carburetion satisfactorily in many different ways is attested by the great variety of commercial carburetors in use. The types vary from simple to extremely complicated, depending upon the degree to which the following requirements are present, and the perfection with which they must be met:

1. Variable power over a range from idling to maximum horsepower at full rated speed.
2. Speed variations from nearly constant speed under governor control to speed controllable from idling to a maximum of several thousand revolutions per minute. Rapid acceleration at varying rates may also be an imposed requirement.
3. Desired thermal efficiency, which may vary from a factor of little importance for some types of engines to one of major importance in others.
4. Required operation when idling may vary from rough and irregular to perfectly smooth.
5. Requirements imposed by the mobility of the engine. If stationary, there is no trouble with surge of fuel in supply chambers. Some engines, as on automobiles, have the carburetors subject to moderate accelerations, but remaining in approximately level position. Aircraft engine carburetors are subject to great accelerations, and those which are employed on acrobatic or combat aircraft may be required to operate in all possible physical positions.
6. The fuels to be carbureted may be of varying volatility.

Most carburetors expect to secure vaporization by the use of a low-pressure spray of fuel (gasoline) from a stationary jet into a rapidly moving stream of air. The purpose of the spray is to expose a large surface of the volatile gasoline for auto-evaporation. As the heat evaporated must come mainly from the fluids themselves, the carbureted mixture undergoes a cooling of several degrees. Other carburetors employ a moderate pressure spray and are adapt-

able to less volatile fuels than gasoline. Still other carburetors have been devised which, instead of a spray, expose a liquid surface in a pool over which is swept the air which it is desired to carburete. Such have been called puddle carburetors, but are deemed to be suitable to meet the requirements mentioned in the preceding paragraph only in the most limited fashion.

A survey of carburetor types may well revert, for background, to the equation for the air-fuel mixture supplied by a simple, low-pressure jet, Venturi carburetor. The equation, as set forth in the preceding section, shows a "coefficient of discharge" ratio C_a/C_g multiplying orifice areas and densities of the air and gasoline. When a simple Venturi carburetor is operated at increasing rates of air flow through it, the numerical magnitude of the coefficient ratio tends, by natural phenomena, to decrease, thus increasing the richness

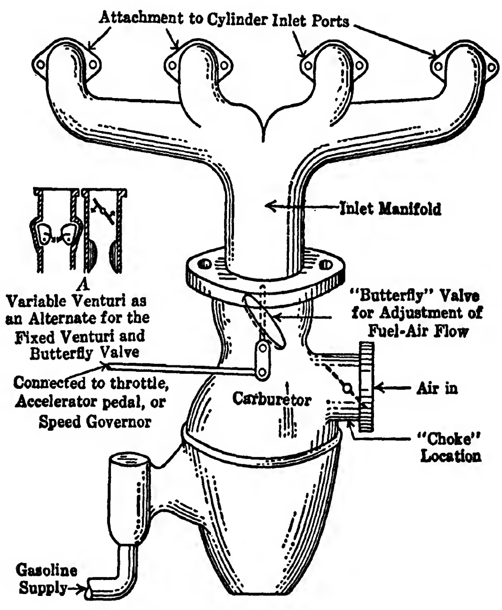


FIG. 8-11. Elements of the carburetor induction system.

of the mixture. To offset this, and obtain approximately uniform mixture under variable air flow, A_a might be increased, or A_g might be decreased. There can be little control over the density of the air, but the effective density of gasoline from the jet is controlled in some types of carburetors by mixing it with variable quantities of air bubbles. Now these three possibilities have all been employed in carburetors. The "variable Venturi" carburetor in effect varies the area of the air orifice. The "metering pin" carburetor is arranged to produce a variable gasoline discharge orifice. The "restricted air bled jet" produces an effect analogous to a change of density of the gasoline. Other compensating principles employed in carburetors involve ingenious uses

of extra jets to supplement the main jet, while others have been designed with auxiliary air valves arranged to admit air to the over-rich carbureted mixture, and so dilute it approximately to the desired richness.

The principal parts of a carburetor might be functionally classified as:

1. A device to meter the fuel into variable air flow to obtain the desired fuel-air ratio.
2. A regulated pressure supply of fuel to the metering equipment.
3. A means for varying the flow of the air-fuel mixture to meet variable demand for power.
4. Auxiliary devices to provide the additional perfection of smooth idling, temporary richness for acceleration, and mixture adjustment to compensate for variable air density occurring during large variations in altitude (aircraft carburetors).

Most carburetors use either the "butterfly" valve or a variable Venturi to impose a variable pressure drop between the carburetor and cylinders for the

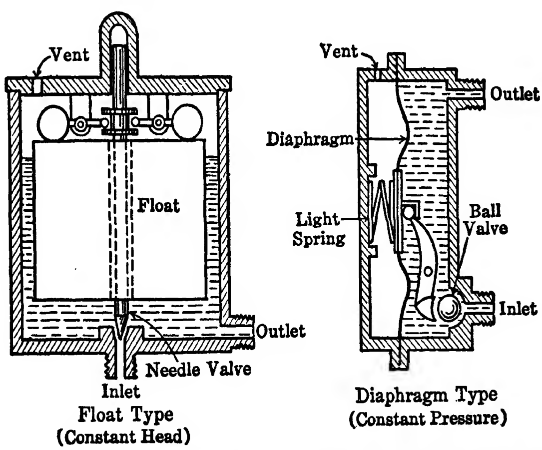


FIG. 8-12. Methods of regulating gasoline supply to metering devices.

purpose of controlling power or speed. To secure a super-rich mixture while cranking the engine at low speed preparatory to starting, the air entrance into the carburetor is almost completely closed by the operation of a valve similar to the butterfly, which is called a "choke." This imposes a high vacuum internally in the carburetor, and assists in drawing large quantities of gasoline violently from the jet. This is needed during starting, because of the quantity of condensation on the cold manifold, and the poor vaporization occurring during starting.

The task of metering gasoline into the air precisely is difficult, at best, and for its success, the metering equipment must be supplied with the gasoline

under constant head or pressure. Between the original source, such as a gasoline supply tank, and the jets, there must be imposed a means for regulating the gasoline supply to the metering devices. Many carburetors use a float-controlled valve arranged to maintain a steady liquid level in a float chamber. This arrangement is subject, however, to the fault of surge and splash, and disruption of the desired level if the carburetor is not maintained approximately horizontal, or if it is subject to severe acceleration. Float control is quite successful for automobiles, trucks, stationary engines, and many other applications, including even aircraft engines where the required operating conditions are not excessively severe. Where the float control is unsatisfactory, constant pressure supply is possible by using a spring-loaded diaphragm whose motion will control the valve admitting fuel from the source of supply. Both of these methods of regulating the supply chamber are illustrated in Figure 8–12.

The success of any carburetor depends on how good its metering system is. There has been previous mention of methods of metering. Some of these will be described in more detail by reference to Figure 8–13. The auxiliary air valve type displays a light spring-loaded valve opening into the carburetor body on the discharge side of the Venturi. The simple jet as shown will tend to produce too rich a mixture at the higher air velocities through the carburetor. These high velocities are accompanied by lower pressures in the region around the auxiliary air valve, which is thereby induced to open to an extent roughly proportional to the decrease of pressure, admitting uncarbureted air and effecting a dilution of the mixture. The variable Venturi carburetor employs a Venturi section (of somewhat imperfect design) whose sides are built of springy strips, or reeds. At slower air speeds the reeds are nearly closed, producing a high vacuum, and improving the tendency of the jet to spray gasoline. At higher air speeds the reeds open more and provide a larger area through which the air can flow. As there is no increase in gasoline jet area, this action is seen to be in opposition to the normal tendency of a static Venturi carburetor to provide a rich mixture at high speeds. Another type is the compound jet, sometimes referred to as the unrestricted air bled jet. Two jets are shown, a main jet M, and an auxiliary jet A. The main jet is a plain jet, and will tend to give an increasingly rich mixture as air flow increases through the Venturi. However, the size of the main jet is not sufficient to produce enough gasoline properly to charge the air which passes the Venturi and needs the action of the auxiliary jet to provide the difference. This auxiliary jet has an inverted characteristic, that is, it tends to give an increasingly lean mixture as the air flow increases. By a suitable combination of the areas of the two jets with the area of the Venturi, an approximately uniform mixture may be maintained over a range of air speed. The inverted characteristic is secured by feeding jet A from a well W which receives gasoline from the

supply chamber through an orifice O. The higher the rate of gasoline flow through A, the more of the available driving pressure that will be consumed by the orifice O. Consequently, the discharge of gasoline from jet A will not increase proportionately faster than the air speed, as is the case with M, but

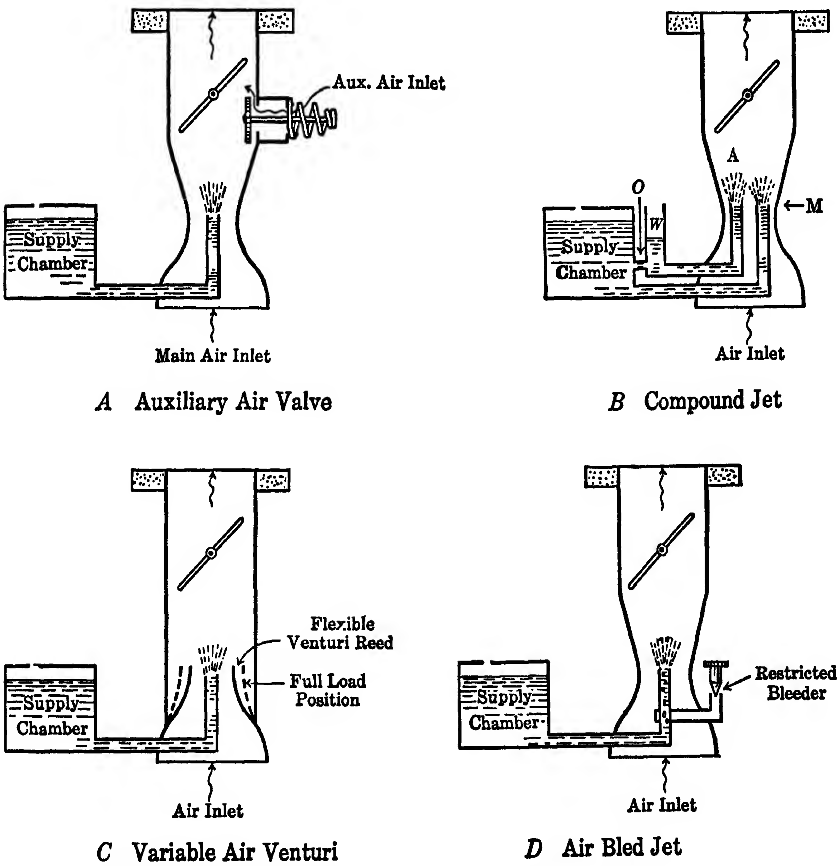


FIG. 8-13. Methods of metering.

will even fail to keep pace with the air flow. Thus, at low air speeds, the gasoline level in W will be nearly up to the constant level of the supply chamber, but will lower as air speed increases, and finally, at maximum speeds, will uncover the orifice, and air will then mix with the gasoline going to the jet, greatly reducing its delivery, and thus compensating for the increase of the main jet.

A jet modifying action of considerable success is illustrated by the air bled jet. This might be termed a restricted air bled jet, and its principle of operation is as follows. A plain jet, opening into the throat of the Venturi, has a side entrance brought into it somewhere near the tip of the jet through

which air may inflow, when the throat pressure is reduced below atmospheric, thus providing a mixture of air bubbles and gasoline in the ascending flow to the jet. Space occupied by the air bubbles cannot, of course, be occupied by the gasoline also, and hence the effect of the air bleeding is to choke the quantity of gasoline that can be discharged from the jet. As the air velocity in the Venturi increases, and the pressure driving the gasoline from the supply chamber up the jet thereby is increased, the induction effect of the air through the bleeder is likewise increased. Suitable proportioning of the gasoline jet to the Venturi orifice, coupled with an adjustment of the air bleeder, has been found to effect very good control on the mixture over a wide range of air flow.

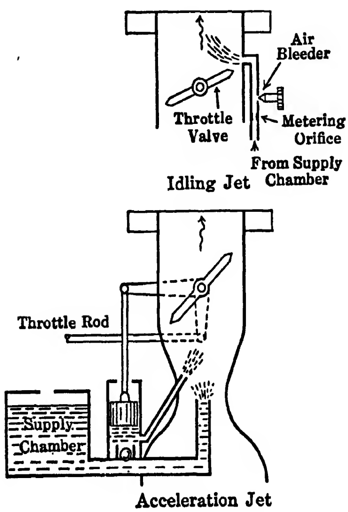


FIG. 8-14. Auxiliary jets.

These principles have been diagrammed in a very elementary way, and without reference to any particular manufacturer's practice. None of these metering principles can effect precisely the desired performance unless they are assisted by other means such as special auxiliary jets, orifices, wells, and the like. To examine in detail these various modifications is beyond our present scope; however, the principles of two common auxiliaries are pictured in Figure 8-14. When idling at very low engine speeds, the throttle valve of an engine is nearly closed, and the flow of air through the Venturi is so slow that no adequate or reliable jetting pressure may be expected. An idling jet is led to an outlet point on the cylinder side of the closed throttle valve, where the suction vacuums are high. The suction will draw up gasoline from the supply to this jet, and a mixture for smooth idling may be obtained by adjusting air bleed into this jet. If the throttle is opened, the suction in this neighborhood decreases, and the idling jet goes out of action. In order to meet the requirements of temporary enrichment of mixture for rapid acceleration, most carburetors, designed for applications where this requirement must be met, are provided with some sort of acceleration jet. The operation of the throttle rod may be used to cause a piston to descend in the accelerating well, forcing an extra supply of gasoline from the accelerating jet. The piston fits loosely in the well or cylinder, and has little tendency to pump gasoline from the accelerating jet if the position of the throttle is changed slowly and smoothly, but the more rapidly the throttle is opened, the higher the pressure set up in the accelerating jet, and the more the temporary enrichment of the mixture.

Thus far, all carburetor principles described have incorporated vertical

flow of the air upward, and open top gasoline jets. The implied position of the carburetor, then, as is shown in Figure 8–11, is below the manifold. Some definite advantages are secured, both in arrangement of the manifolds, in the carburetion, and in the location of auxiliaries about the engine if the carburetor can be placed above the manifold, with the air sweeping downward through it, instead of upward. Such carburetors are called *downdraft.* The open top jets obviously are not suitable for the downdraft carburetor, but no great difficulty is met in carbureting horizontally out of jets. Any or all of the metering modifications heretofore discussed may be applied in downdraft carburetion.

The reader is asked to refer once again to the equation for mixture, and note that if all other terms in the equation were constant, the mixture would again become too rich, because of the decrease of air density, if an engine were employed at increasingly greater altitudes above sea level. Surface vehicles powered by gasoline engines may have their carburetors adjusted from time to time for operation in regions of different altitude, usually by needle valve adjustments which impose orifice-like restrictions in the flow to the main jet. Aircraft may travel into regions of widely varying air density frequently and without the possibility of a static adjustment to compensate for variable air density. Thus, carburetors which are installed on aircraft capable of attaining altitudes where the change of air density would affect the carburetion in a detrimental way, are equipped with *mixture control* which may be operated manually or may be automatically adjusted by the atmospheric pressure itself. This mixture control is usually either of the needle valve or back suction type. In needle valve adjustment, what amounts to a variable orifice is imposed in the flow line leading to the gasoline jet, and the setting of this orifice may be adjusted by the engine operator through a mechanical linkage leading to the carburetor. Alternately, the needle valve adjustment may be obtained automatically by spring-loaded bellows whose expansion is a reflection of the existing air pressure, and hence indirectly of the air density. It will be seen in the various figures of this section that the gasoline supply chamber is vented to the atmosphere. If it could be arranged partially to close this vent, and connect the region above the liquid level with the throat of the Venturi, the normal pressure differential delivering gasoline through the jet could be interfered with, and, in fact, completely dispersed at will. By regulating the degree of interference through adjustments of the vent opening (and possibly also the valving of the connection to the Venturi), the jetting pressure could be cut back and a compensation effected for the decreasing density of the air at higher altitudes. This is called back suction control.

The problem in securing good carburetion reaches its maximum difficulty in the supply to airplane engines required to engage in acrobatic maneuvers and operate at high altitudes as, for example, fighter airplanes. As has been

mentioned, float control of supply would have to be replaced by some other system, also vapor lock and ice formation might occur in a conventional carburetor at high altitudes. These difficulties will be overcome by injection carburetion, and, in addition, it might be possible to use fuels of lower volatility. A direct solution of this problem would be to inject metered amounts of fuel directly into the cylinder by injection valves. This relieves the manifold of fuel distribution problems but distributes the fuel system about the engine, whereas there are advantages of keeping the fuel supply system as compact as possible. An injection carburetor which may be a single unit, and thus obtain the simplicity and compactness of the float carburetor, has been built and is in use on military airplanes. This injection carburetor injects a metered spray of fuel into the air inlet at about 5 psi. gage. This is sufficient to atomize the gasoline finely, and mix it with the air, and the carburetion will be completed while the mixture flows through the supercharger and inlet manifolds into the cylinders. The carburetor is floatless, has automatic mixture control, and allows the power of the engine to be controlled by a throttle rod exactly as in the case of the float carburetor.

8–9. Spark Plug. The spark plug is a device inserted in the combustion chamber of spark ignition engines to provide the insulated electrode and the gap necessary in the high-tension jump spark ignition system. As the electrodes are oxidized and pitted by usage, and since insulators may suffer damage, or carbon and oil may accumulate to cause short-circuit, the firing points should be made removable for adjustment, cleaning, or replacement. Universally a threaded hole is provided through the wall of the combustion chamber and the ignition points are built into a small compact "plug" which is provided with a threaded steel shank so that the plug may be screwed into the combustion chamber. Only one of the electrodes leading to the gap needs to be insulated since the electrical circuit is completed through the plug shell and engine body by grounding one end of the high-voltage supply. In operation, parts of the plug remain at high temperatures. The electrodes are made of special alloy steel suitable for resisting the high temperature and the insulator must be capable of withstanding both high temperature and electric stress of the order of several thousand volts. Molded porcelain and built-up stacks of mica rings are used.

Satisfactory operation of the engine is dependent upon proper spark plug temperature. Plugs can be designed with longer or shorter travel of the heat from the interior tip of the insulator to its point of contact with the shell, resulting in higher or lower plug operating temperatures. If the plug is too cool, carbonizing and short-circuiting of the gap will be probable; if too hot it may promote pre-ignition and detonation. The length of the gap affects the energy of the spark and the voltage required to produce it. Small gaps reduce electrical stress, make starting easier, but are more easily fouled by bits of carbon

and do not ignite the gasoline as efficiently, as has been proved by tests for thermal efficiency under different gap settings. About .03 in. ($\pm$.01) covers the recommendations for plug gaps for all engines except aeronautical, where the gaps are much smaller because of advantages of limiting voltage stress on the plugs and ignition wiring at high altitudes.

A greater variety of spark plugs has been produced for aeronautical service than any other because service conditions are more severe and reliability more essential than in other internal combustion engine fields. Also the use of radio receiving equipment is an important adjunct to navigation and the ignition

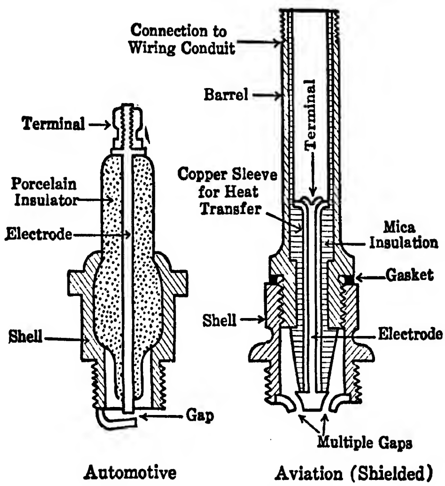


FIG. 8-15. Spark plugs.

system, including spark plugs, will interfere with radio reception unless properly shielded. Operation of aircraft at extremely high altitudes poses further problems in corona loss from the high-voltage portions of the ignition system. Electrically shielded and gas-pressurized plugs and ignition wiring harness are employed to meet these special problems of aircraft ignition.

8-10. Ignition System. Back of the spark plug is an electrical system which must generate, and time to a split second, the voltage pulse which flashes the spark through the compressed explosive mixture between the electrodes of the plug. This ignition system must produce a focal point of heat sufficient to ignite the mixture, and to provide it at exactly the right instant. The average time allowable in a gasoline engine operating at about 2000 rpm. for ignition and explosion of the mixture following ignition is only about one one-hundredth of a second. Furthermore, it has been found necessary to vary the timing of ignition with respect to the cycle of operation in such a way that ignition occurs relatively earlier in the cycle the higher the rotative speed. These conditions are well met by the electrical spark system of ignition, in

which a high-voltage spark jumps across a stationary spark gap. Due to the high speed of electrical impulses in wires, there is no difficulty in accurately timing the ignition in an electrical system. However, the high-voltage jump spark method of ignition is not the only electrical form in use.

A low-tension ignition system is one in which the voltage at the ignition points is insufficient to cause a spark to jump a static gap. The electrodes of the gap are brought together, and then rapidly separated. In this way, a low-voltage supply is able to cause an electric arc to follow the separation of electrodes. Figure 8–16 shows a system of this nature. It consists of a source of voltage (a battery), an induction coil, and ignition points. The ignition points and their operating mechanism constitute what is known as the igniter. The igniter is mounted in the combustion chamber, and is composed of one stationary, and one moving electrode. The moving member is mechanically operated from the crankshaft by means of a shaft extending through the cylinder wall, and in this way the igniter is synchronized with the cycle. When the igniter points are pushed together, making contact, an electric current flows through the coil, igniter points, and battery. If, then, the current is suddenly interrupted through rapid opening of the igniter points, the battery voltage, aided by self-induced voltage of the induction coil, will cause an arc to follow. This arc is the source of ignition. This low-tension system is only occasionally used on engines, usually on relatively slow-moving, fixed, or semi-portable engines.

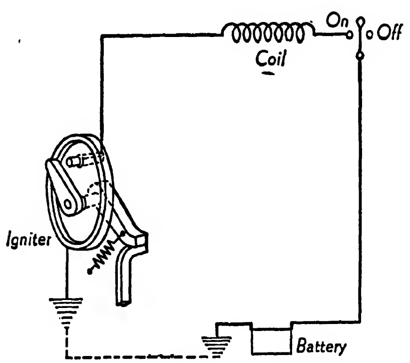


FIG. 8–16. Low tension system for single cylinder stationary engine.

The high-tension system, shown by Figure 8–17, is the system used on most gasoline and gas engines, and is familiar to many persons through its use as the ignition system of the automobile engine. It is well adapted to multi-cylinder engines, and is especially suitable where other electrical services are required; for example, lighting and electric cranking. Two systems of high-tension ignition are (1) the battery and coil system, which is most popular today on automobiles; (2) the magneto system, which is more widely used on aeronautical and fixed or semi-portable engines.

A source of low voltage, i.e., the battery at 6 volts, is connected to the primary of an induction coil through an automatic switch which is opened at regular intervals by the engine itself. This automatic switch is called the interrupter, timer, or *breaker*. Whenever it opens the circuit, the induction coil generates a high voltage which is sufficient to cause a spark to jump the

gap in the combustion chamber. The use of a single coil to supply high-voltage current to the spark plugs of a multi-cylinder engine involves another mechanically operated device, the distributor. The distributor picks up the high-voltage impulse as it comes from the induction coil, and shunts it to the cylinder ready to receive ignition action. Depending upon the sequence with which high-voltage leads are attached between spark plug and distributor, the engine will operate with a definite sequence of power strokes derived from

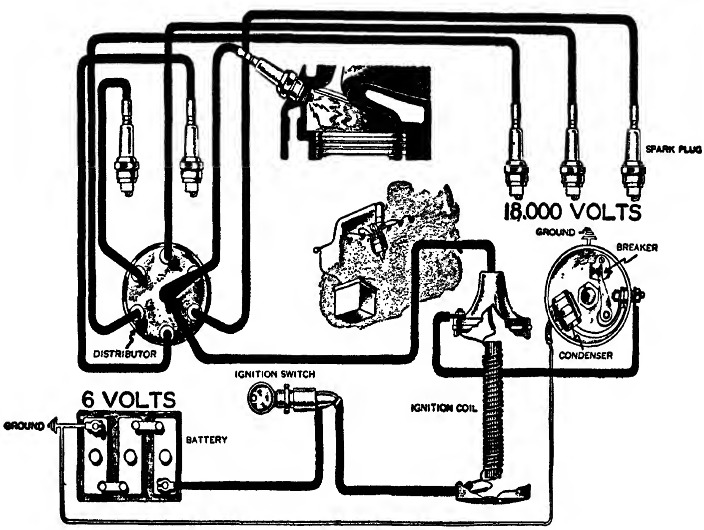


FIG. 8–17. Automobile ignition system. (Courtesy General Motors Corp.)

the different cylinders. There are certain standard arrangements of this sequence for the common multiples of cylinders employed in gasoline engines. These arrangements are called the *firing orders*. The firing order must be well chosen to prevent rocking or galloping of the engine.

In the common four-cycle engine, one ignition impulse serves a cylinder for two complete revolutions. Because of this, the distributor rotates at one-half engine speed. The breaker is usually driven by a cam which does not necessarily have to rotate at one-half crankshaft speed, but which can conveniently be made to do so, since a one-half speed shaft must be provided for the distributor. If the breaker is operated by a cam revolving at one-half crankshaft speed, the cam must have as many lobes on it as there are cylinders in the engine. The adjustment of the instant of ignition in a cycle with due regard to the demands of variable speed operation can be done through a slight rotative displacement of the case which carries the breaker points. Formerly this action was accomplished manually, but now manufacturers have developed automatic spark advances, where centrifugal weights opposed

by springs or diaphragms actuated by intake suction adjust the breaker to a position of spark advance which will be best for the speed of the engine. A condenser paralleled across the interrupter points absorbs energy which would otherwise be manifested as a small arc following the opening of the points. It is convenient to mount this condenser, interrupter, and distributor as a unit enclosed in a single case, and driven by a one-half speed shaft. A unit of this type is shown in Figure 8-18.

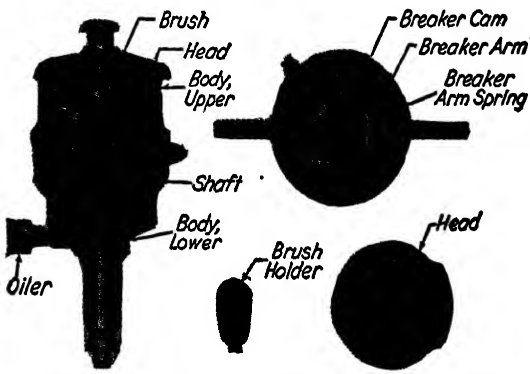


FIG. 8-18. North-East ignition unit.

8-11. Magneto. The magneto is a device for producing alternating currents of high voltage properly synchronized for distribution to the spark plugs. The same function may generally be admirably performed by a coil and battery system, but the compactness and light weight of the magneto, and its reliability, have caused it to be widely used for ignition. The magneto is but seldom seen in that most familiar example of spark ignition, the automobile engine, because the magneto performs but a single function—ignition. Except for spark plugs, wiring harness, and switch, it is a complete ignition system. Nowadays it is to be found on engines such as stationary engines, tractor engines, boat engines, and others where the ignition system is not wanted involved with other electrical services; and on the aircraft engine, where its proven reliability is highly respected.

In reality the magneto is an electric generator, induction coil, breaker, and distributor, all consolidated in one small compact unit. Three different systems of generation of the low-voltage electrical pulses are found. These, which are illustrated in Figure 8-19, may be called (1) the armature type, (2) the inductor type, and (3) the rotating magnet type. The armature and inductor types have stationary permanent magnets. In one, a wire-wound armature is rotated between pole shoes, as in a dynamo machine, and in the other a soft iron "inductor," or flux changer, is rotated between the pole shoes.

The magnetizing influence which a magnet can exert on a magnetic circuit is called magnetomotive force. It is the "voltage" of a magnetic circuit.

Flux in a magnetic circuit is equivalent to current in an electrical circuit. It may be represented by imaginary lines of force, as in Figure 8–19. The stronger the magnetomotive force of a circuit, the greater will be the flux. The material of the magnetic circuit may be composed of anything through

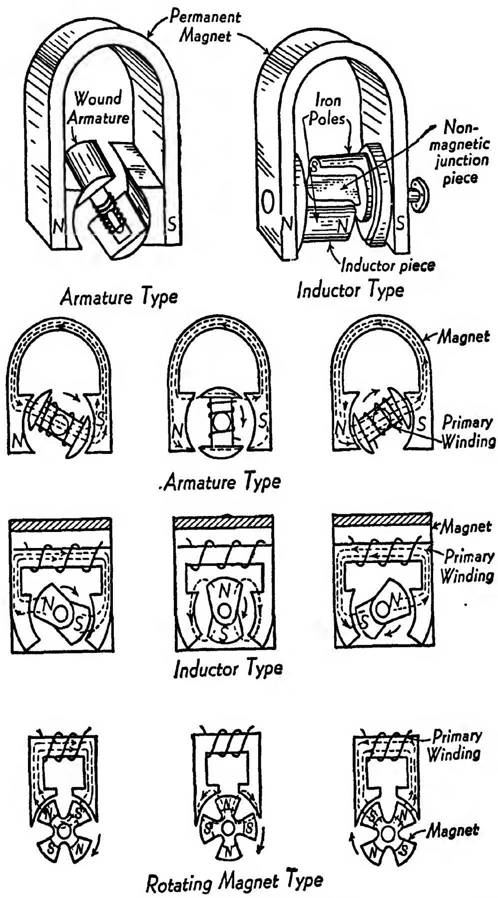


FIG. 8–19. Diagram of flux flow and magnetic field variation.

which the magnet can force flux, even air, but is mainly ferrous because of the excellent magnetic "conductivity" of this material, especially soft iron. Only the smallest of air gaps can be permitted because of the high "reluctance" of such to convey flux. If the amount of flux associated with a coil is varied by any process whatsoever, an electromotive force is induced in the coil. In all magnetos, motion of the rotor induces a change of flux linkage through the primary coil winding. The method of varying flux linkage appears diagrammed in the figure. Flux lines are shown dotted, with assumed direction of flux being from a north to a south pole. The diagrams are self-explana-

tory, although possibly some comment will be helpful in tracing the magnetic circuit of the inductor type. In the figure it is seen that the rotating soft iron inductor has N and S pole pieces which remain magnetically in contact with the poles of the permanent magnet, although continuously rotating. The N and S inductors are magnetically separated by the middle section of the inductor being non-magnetic (brass). Consequently, the flux lines leaving the N pole of the magnet pass through the N end of the inductor, thence to the pole shoe and through the soft iron core of the primary winding to the other pole shoe, then across the air gap into the S end of the inductor, back to the S pole of the permanent magnet and through the magnet to the starting point.

The voltage induced in the primary winding is insufficient to jump a spark across the gaps of the spark plugs, but if a circuit breaker is connected in the coil circuit to open the previously short-circuited coil at the instant the *flux change* reaches maximum (not coincident with maximum flux density), the resulting surges of primary coil current apply extra magnetomotive force to the magnetic circuit. Then if a secondary coil of a great many turns is wound around the primary coil, extremely high voltages will be induced in it. It is these voltages which provide the "pressure" necessary for a jump spark. The secondary coil lead could be connected directly to the spark plug of a single cylinder engine. A four-cylinder, four-cycle engine needs four precisely timed sparks every two revolutions of the crankshaft. Consequently, the magneto rotor speed, the number of poles, and the interrupter design must all be correlated so as to produce the four high-voltage surges. There must, in addition, be a distributor, usually an integral part of the magneto. The distributor must be geared to rotate at one-half of crankshaft speed. The distributor rotor (finger, brush) makes its connection with the magneto secondary through the drive shaft. The distributor case is molded of high-grade insulation with conducting inserts to which are connected the wires which carry the voltage to the plugs. In some distributors the rotor "brushes" the contact points as it passes them, while in others the current must jump a small gap between the moving finger and the stationary contact point.

The primary circuit must be broken as many times per single revolution of the rotor as there are permanent or induced poles on the rotor. The break is mechanically accomplished by a cam. Usually the cam is mounted directly on the rotor shaft. In a four-cycle engine the number of firings required per revolution is one-half the number of cylinders. The required magneto shaft:crankshaft speed ratio is the number of cylinders divided by twice the number of poles on the rotor. Figure 8–20 shows typical breaker points. The movable point is mounted on a spring. Points are made of platinum iridium alloy, and have a condenser connected across them to eliminate arcing and burning from self-induction in the primary circuit, and also to assist

in bringing primary current to zero rapidly, thereby increasing the voltage produced by the secondary winding.

Illustrating a standard type magneto for six-cylinder engine ignition, Figure 8–21 shows an external appearance and a section through a rotating magnet type of magneto. As the auxiliary view consisting of a circuit diagram explains, the two-pole rotor rotates between pole shoes A and B in order to vary flux linkage through the coil.

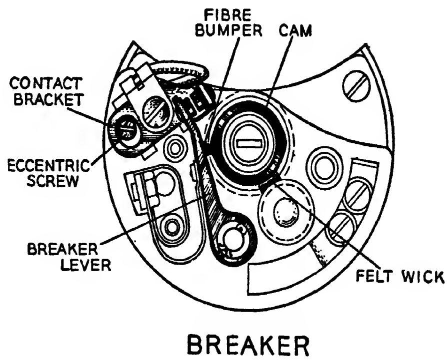


Fig. 8–20. Breaker points.

Magnetos are regularly employed in modern aircraft engine ignition. That field of use poses operating problems not encountered elsewhere, some of which are worthy of description. Since inductor and rotating magnet types have no high-voltage lead-off from the rotor, they are favored over the armature (shuttle) type. A typical magneto system is pictured in Figure 8–22. Two

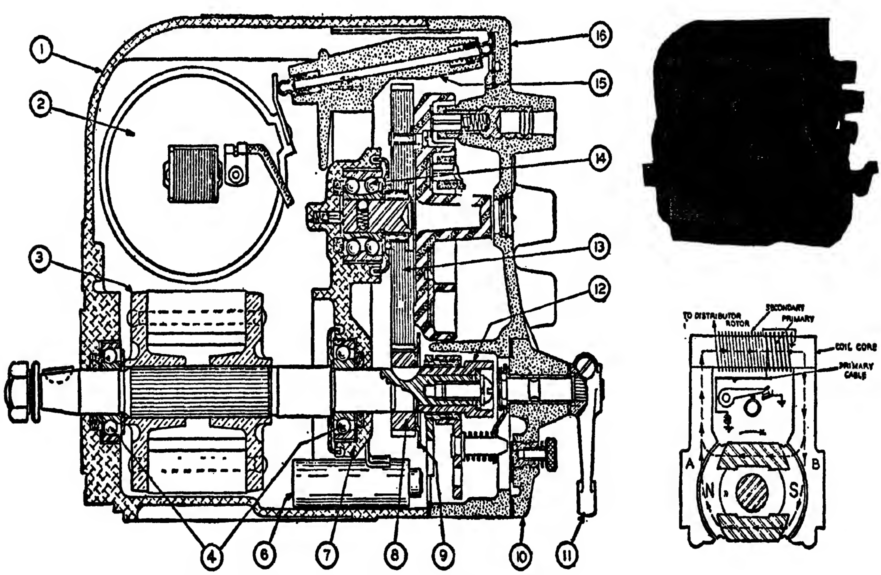


Fig. 8–21. High tension ignition magneto. 1. Frame. 2. Coil. 3. Rotor. 6. Condenser. 8, 13. Gears. 11. Advance lever. 12. Breaker cam. 15. High tension lead. 16. Distributor case. (Courtesy American Bosch Corp.)

magnetos are provided for the engine (9 cylinder, radial, indicated by the wiring harness) so that two independent sources of ignition will exist. Each cylinder has two spark plugs, each plug connected separately to a dif-

ferent magneto. The magnetos are mounted on the engine crankcase and gear driven from it. The ignition switch which is at the pilot's control is turned to short circuit the breakers in order to stop the engine. In some cases, during cranking of the engine the magneto rotor speed is too slow to produce a satisfactory spark. A supplementary high-voltage source known as a booster may be used during cranking, or an impulse coupling of the magneto rotor

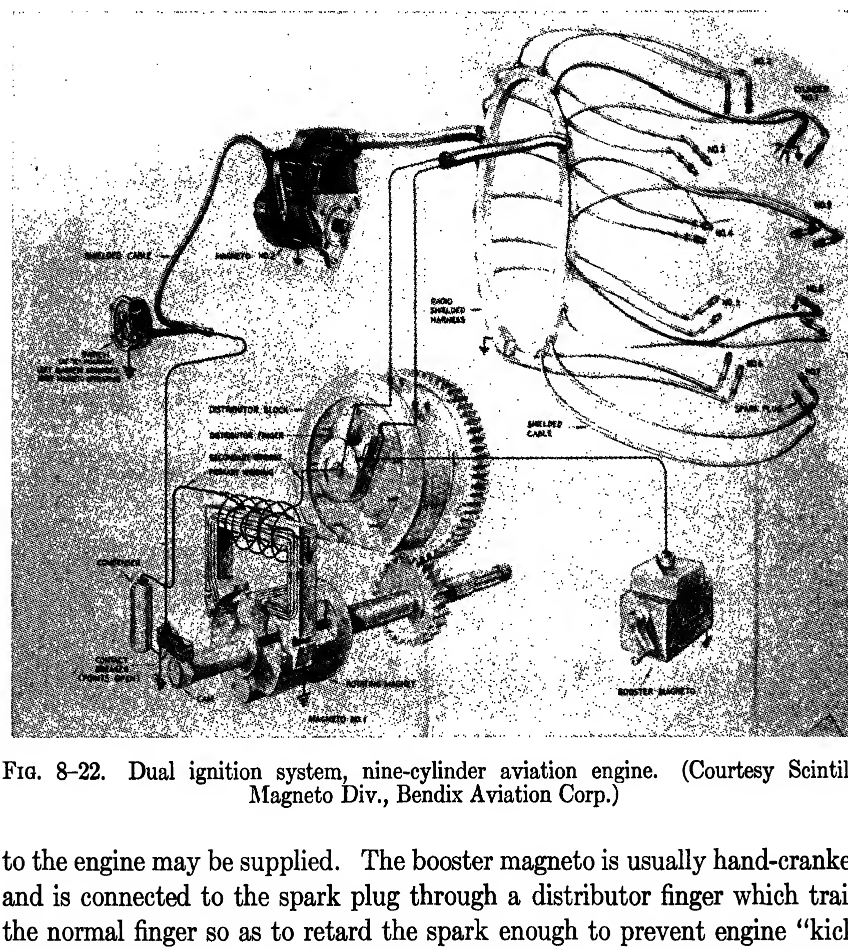


Fig. 8-22. Dual ignition system, nine-cylinder aviation engine. (Courtesy Scintilla Magneto Div., Bendix Aviation Corp.)

to the engine may be supplied. The booster magneto is usually hand-cranked and is connected to the spark plug through a distributor finger which trails the normal finger so as to retard the spark enough to prevent engine "kick-back" during cranking. The impulse coupling is a spring-like mechanical linkage between the engine and magneto shafts which will wind up and let go at just the right moment to spin the magneto shaft fast enough to generate the necessary spark voltage. After the engine has started and reached a predetermined speed where normal magneto action is adequate, a centrifugal weight-actuated lock renders the impulse device inoperative. The ignition lag necessary to reduce kickback of an engine which normally operates on about 20° advance of spark before top dead center can be built into the dis-

tributor finger design used with a booster magneto (or vibrating coil and battery), or into the impulse coupling. The case on which the breaker is mounted, if permitted some small angular adjustment, can become a means of retarding the spark, for if the breaker shaft were rotating at crankshaft speed, a five-degree rotation of the case in the direction of rotation of the cam would delay the spark 5 degrees of crankshaft travel. Some magnetos have this spark-retarding feature, as do the breakers on most all coil and battery systems, but it is uncommon in aircraft ignition.

The ignition system of aviation engines could be much like that for any other service, except for the following service conditions:

1. The growing importance of radio communication to navigation and piloting requires special attention to prevent broadcasting of interference static from the ignition system.
2. Increasing demands for high altitude equipment have accentuated the effect of corona loss, ionization flashover, etc.
3. The need in high-power engines for a system that will fire successfully even though the normal insulation throughout the ignition system is greatly impaired.

8–12. Cooling of Internal Combustion Engines. The products of combustion in an engine are necessarily at a high temperature when formed. If the cylinder walls are not maintained well below 500°, overheating and breakdown of the piston-cylinder lubrication film can ensue. A deliberate and purposeful cooling of the outside of the cylinder wall is imperative. Usually about a third of the heating value of the fuel consumed must be abstracted by a system of cooling which therefore becomes an essential and major element of any internal combustion engine.

The heat received by the cooling system is dissipated to the atmosphere in the case of land mobile units such as automobiles, railcars, etc.; to the water on which engine-propelled vessels float; and either to the atmosphere or to large natural bodies of water or flowing streams in the case of stationary engines. Cooling systems may be divided into (1) those using air directly, and (2) systems employing a liquid for the cooling of the cylinder and combustion chamber. Liquid cooling is accomplished by surrounding the cylinder with a cooling jacket containing the liquid. Water is ordinarily employed, but there are two exceptions. One is met where cooling systems of an idle engine are exposed to water-freezing temperatures. Liquids of lower freezing point, either pure or in solution with water, are then substituted for water. Alcohol and ethylene glycol are two such liquids. There is another reason for using the latter as a cooling system liquid. Its boiling temperature is about 380° F, therefore the jacket liquid may be heated to a possible 250° instead of

the 180° common in circulating water systems. This materially reduces both the required flow of coolant and the size of the radiator.

Direct air cooling is obtained by circulating air over the exterior of the cylinder. Heat transfer from cylinder to air is so much slower than to water that it becomes necessary to extend the exposure of metal surface by casting or machining cooling fins on the exterior of the cylinder and combustion chamber. Air- and liquid-cooled cylinders are compared in Figure 8–23.

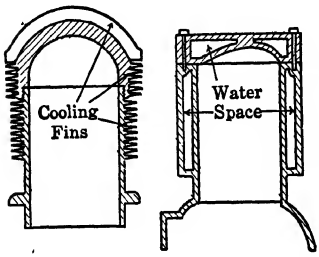


Fig. 8–23. Air- and liquid-cooled cylinders compared. (Valves and plugs omitted.)

Liquid-cooled systems may be further subdivided into evaporative and circulating types. Evaporative cooling is found on small stationary engines having *hopper cooling*. The water jacket is extended into an open hopper, or reservoir, so that a considerable quantity of water may be held. During operation of the engine the water is boiling. Heat is convected away from the engine in the form of steam. Each pound of water evaporated will remove as much heat as 15 or 20 lbs. of water circulated, but, unlike a circulating re-cooling system, water must be added periodically to the hopper to replace the evaporation. Most liquid-cooled engines employ the circulating system wherein the liquid is cooled externally and readmitted to the engine. A steady flow is maintained through the cooling jackets.

Methods of cooling the liquid are:

1. Pass the liquid through a "radiator" where by conduction and convection (but very little radiation) heat may be removed by air blown through or across the radiator.
2. Pass the liquid through a surface heat exchanger and absorb the heat in raw water which may then be re-cooled or wasted.
3. Pass the liquid through a cooling tower or spray pond to cool it evaporatively.

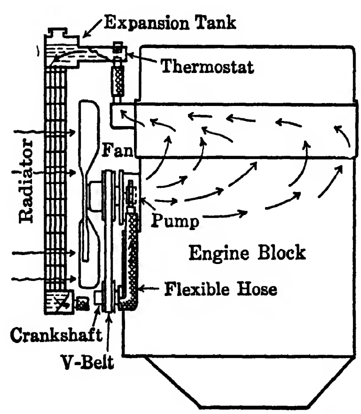


Fig. 8–24. Forced circulation water cooling system.

A typical water cooling system for a gasoline tractor engine is shown in Figure 8–24. Water is circulated from the engine jackets to a radiator core where it is cooled, then withdrawn by an engine-driven centrifugal pump and sent through the cylinder jackets. From there it flows into the jackets surrounding the cylinder head and then back to the radiator. Since the radiator must be of sufficient

size to obtain adequate cooling in hot weather, a thermostat is inserted in the flow line to maintain suitable engine temperature in cold weather.

Direct air-cooled engines are used in many power fields. Some have blowers to force the air to circulate over the cooling fins, whereas others depend on the motion of the engine itself to create the circulation. A motorcycle engine is illustrative of the self-cooled principle. The airplane engine is another. However, the latter is essentially a high output engine installed under conditions demanding utmost efficiency of cooling. The air is carefully guided over its cooling fins by *cowling* so that the air used is the least possible

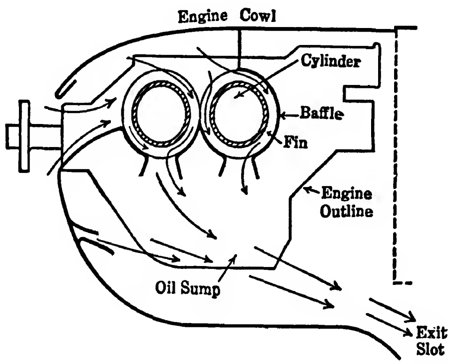


FIG. 8–25. Cowling and pressure baffles for horizontally opposed airplane engine.

and its usage is accomplished with a minimum of disturbance to the surrounding air stream. Effective cooling of most air-cooled engines is so dependent on the cowling that it cannot be described except in conjunction with the cowling. Cowling is removable covering placed over or around an engine. Its purpose may be simply that of a weather guard—or it may serve the more complex function of streamlining or engine cooling. The *hood* of an automobile engine is a simple form of cowling. It is in connection with air-cooled airplane engines that cowling is of the greatest technical interest, since there it becomes part of high performance design. Air-cooled engines habitually operate at higher temperatures than liquid-cooled, which gives them some advantages.

The amount of heat to be removed by the cooling system is related to engine operation variables as follows:

Increase of engine power increases required cooling.
Increase of air fuel ratio decreases required cooling.
Increase of combustion air temperature increases required cooling.

Example 1: A hopper-cooled engine was operated at constant power for a period of time during which 4.1 lbs. of gasoline were consumed, and 2.3 gals. of water were added to keep the water jackets full. Find the per cent of the lower heating value lost to cooling.

Assuming that the hopper water was hot at the beginning of the test period, the make-up could be considered to have been converted from 60° F to dry saturated steam at atmospheric pressure by the cooling losses.

$$\text{Cooling losses} = 2.3 \times 8.33(1150.4 - 28) = 21{,}500 \text{ B.t.u.}$$

$$\text{Percentage of cooling loss} = \frac{21{,}500}{4.1 \times 19{,}500} = 26.9\%.$$

Example 2: How much water, which is being externally cooled from 180° F to 130° F, must be circulated per min. in the cooling system of an 85-hp. engine? Consider that thermal efficiency is 26% and cooling loss will be 30%.

$$\text{The output per hr.} = 85 \times 2545 = 217{,}000 \text{ B.t.u.}$$

$$\text{Cooling loss per min.} = \frac{217{,}000}{60} \times \frac{.30}{.26} = 4170 \text{ B.t.u.}$$

$$\text{Lbs. water circulating per min.} = \frac{4170}{180 - 130} = 83 \text{ lbs. (10 gals. per min.).}$$

8–13. Lubrication. The lubrication of this type of engine is of the utmost importance because of high speed, temperature, and pressure, coupled with the interdependence of all parts upon the proper functioning of the remainder. Parts needing lubrication are the piston and cylinder, the valve gear, the connecting rod bearings, the crankshaft, and camshaft bearings. Accessories such as fans, generators, starters, also have their points of lubrication.

The three systems of lubrication most used are the splash, the semi-pressure feed, and the pressure feed, all of these relating to the enclosed crankcase type engine. In the splash system oil is pumped into troughs located so that the ends of the connecting rods dip into a pool of oil and splash it about inside the crankcase. The semi-pressure system has lubrication of the crankshaft bearings and the camshaft bearings by oil which is conveyed to them under pressure through tubes. The connecting rod bearings and cylinders are lubricated by splash. There is neither trough nor splash in full pressure feed. Oil from the pump is carried to the crankshaft bearings through tubes, and to the connecting rod bearings through drilled passages in the webs of the crank. Centrifugal force throws it out of the connecting rod bearing as the latter revolves, so that the crankcase is filled with an oil mist which lubricates the piston and wrist pin. Occasionally one will find the connecting rod drilled in its entire length so that the piston end of the connecting rod also receives oil under pressure. The oil so delivered to the bearings drains into the crankcase, from which it is either taken as needed by the oil pump, or is immediately removed and placed in an external tank by a scavenging oil pump. These two systems are known as the *dry* and the *wet sump* systems.

During the operation of the engine there is a steady loss of oil. Some is burned in the combustion chamber, some leaks through gaskets and around

shafts, and some is blown out of the crankcase breather as a mist. That which is burned in the combustion chamber accounts largely for the accumulation of carbon so often found there. Some oil in the combustion chamber is necessary if the piston is to be properly lubricated, but wear or sticking of the piston rings may cause the piston to assume some pumping action, and deliver to the combustion chamber a great deal more oil than would be necessary for cylinder lubrication, certainly more than is desirable from the standpoint of the condition of the combustion chamber. In addition to creating a carbon

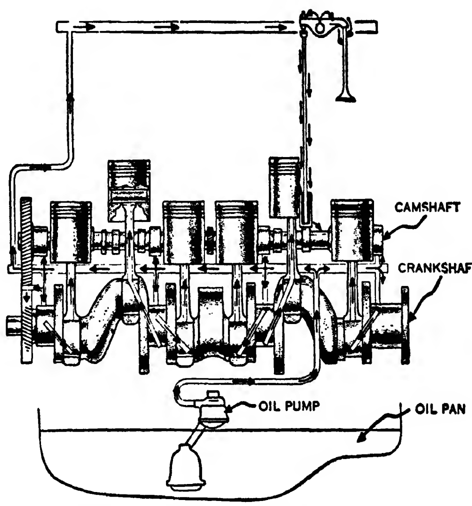


FIG. 8-26. Oil pressure system. (Courtesy General Motors Corp.)

deposit, an excessive amount of oil may foul the spark plug points, and cause sticking of the valves. Carbon in the cylinder heads provides a black surface which initially aids heat transfer, but, as it thickens, greatly retards heat transfer, due to the low heat transfer coefficient. Then hot spots remain in the cylinder and pre-ignition and detonation are liable to occur.

When an excessive quantity of oil passes the piston the defect can be remedied sometimes by removing and cleaning the rings and their grooves. But if mechanical wear is the cause, new oversize pistons and/or rings will be required. It may become necessary also to recondition the cylinder by reboring and honing to slightly larger diameter, or by replacing the liner.

8-14. Starter. The cycle of the internal combustion engine requires that the engine be initially revolved from an external source of power. The mechanism supplying this initial effort is known as the starter. Certain small semi-portable or stationary engines, also tractor engines, truck engines, etc., are frequently started by manual cranking.

Starters for automotive engines are generally electric motors of a series wound type, deriving electric current, when needed, from a storage battery. The relation of motor demand to battery capacity is such that the use of the motor must be restricted to short intervals of time, between which the battery is recharged. Normally, the starting motor is disengaged from the engine. During the starting cycle it is connected by a reduction gear which multiplies the torque produced by the motor.

Aeronautical engines can be started by swinging the propeller by hand. However, many are now equipped with starters, the commercial plane because of the size of the engine and the inconvenient location of the propeller from the standpoint of manual starting, the small plane for convenience of the private owner, or in emulation of the standard practice on automobiles.

8–15. Capacity and Performance. The performance of any prime mover is of interest to the owner, operator, and designer, as is also the variation in performance created by changes in mode of operation. While many features of prime mover action might be designated as performance, the items usually selected are: (1) power output; (2) thermal efficiency; (3) speed, usually rotative.

The output of an engine is its "shaft horsepower," sometimes called *brake horsepower*. Since the purpose of obtaining and using an engine is for its power, the brake horsepower is of first concern in any event. The operating expense of an engine consists mainly of the cost of fuel; hence fuel consumption (i.e., quantity per unit time at a specified output), specific fuel consumption (lb. per brake hp.-hr.), or thermal efficiency are important, being measures of the direct cost of producing power. Rotative speed, say revolutions per minute, affects size and weight of the engine; also, its application to specific driven machinery, since direct drive, if possible, is usually simplest and best.

The internal combustion engine combines fuel and air in combustion for the production of a gas pressure against the piston. A basic factor of performance is, therefore, atmospheric air. But this may vary greatly both in temperature and density, especially in the aeronautical field. To combat loss of power due to diminishing atmospheric density at altitudes, airplane engines are often provided with superchargers to compress that air toward sea-level density. However, should this compression also be applied to sea-level air, the resulting charge would be so powerful as to damage the engine by overstraining and overheating cylinders, pistons, bearings, etc. It is necessary to throttle the intake so that the cylinder mean effective pressure will be limited to a safe value. This adds *manifold pressure* to other factors affecting power output.

Because friction and cooling heat losses tend to remain constant though output varies, thermal efficiency is not constant but decreases at part capac-

ity. Correspondingly, the specific fuel consumption increases. This variation (with constant rpm.) is visualized in Figure 8–27, part load being carried by throttling the inlet so as to reduce the fuel admitted per cycle. If the inlet is unrestricted, but the output is varied by adjustments of the connected load, so that speed increases, the output will first rise, then decrease as a speed is reached where volumetric efficiency is decreasing more rapidly than number of power cycles increases. Likewise, fuel consumption is increased at slow speeds because more heat is transferred to the cooling system as a result of the relatively longer time the hot gases are exposed to the cylinder walls.

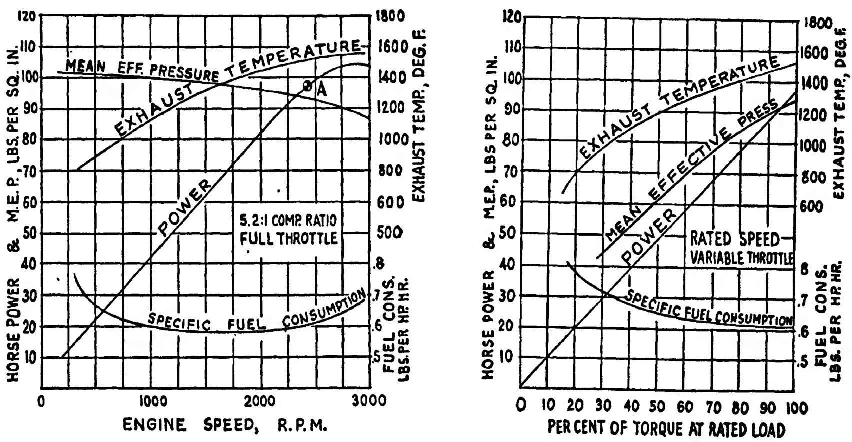


FIG. 8–27. Typical S.I. Engine Performance. Point *A* represents the arbitrarily selected "rating." The less conservatively the engine is rated, the nearer point *A* will be to the peak of the power curve.

Again, it increases at high speeds because of increasing mechanical friction. Some intermediate speed will exhibit an optimum combination of cooling and friction, producing minimum fuel consumption and maximum thermal efficiency.

Ruggedness, reliability, long life are features that cannot be expressed by the factors shown in these figures. High speeds can result in damaging centrifugal and reciprocating stresses. Excessive supercharging of engine cylinders forces quantities of fuel and air into the combustion chamber, where excessive pressures and temperatures can be produced.

Cylinder mean effective pressure, or the hypothetical *Brake Mean Effective Pressure*,* is controlled by throttling the induction system. The same power may be produced by various combinations of induction manifold pressure and rotative speed lying between the extremes of maximum allowable m.e.p. with low speed and maximum speed with low m.e.p.

* Defined in Section 9–10.

Power is near maximum at rated speed. The number of power strokes per minute in a four-stroke cycle engine of a single-acting type is one-half the number of revolutions per minute times the number of cylinders. In a two-stroke single-acting engine it is the number of revolutions per minute times the number of cylinders. In the double-acting engine, it is always twice that of the corresponding single-acting engine. Thus a two-cycle, double-acting engine would have four times as many power strokes for the same rotative speed as a four-cycle single-acting engine. The amount of power developed by the S.I. engine may be varied in several ways, among which are the following:

1. *Quantity Governing.* In quantity governing, the amount of air mixed with a unit of fuel is kept the same, but the quantity fed per cycle is decreased for decreased power, and vice versa. This system is the type ordinarily employed on the automobile engine. The disadvantage of quantity governing is that it reduces the efficiency because the full compression pressure is not reached when the engine is operating throttled.

2. *Quality Governing.* With this the full compression pressure is reached at the end of the compression stroke because the amount of air charged per cycle remains constant. Power is varied by altering the mixture of fuel and air. This system of governing is faulty in that unexplosive mixtures are produced at very light loads, causing a hit-and-miss type governing. It is suitable to gas, but not gasoline, engines.

3. *Hit-and-Miss Governing.* An inexpensive simple form of constant speed governing used on light portable and stationary engines, is known as hit-and-miss governing. It is governing by means of varying the number of power strokes per minute by causing some normal power strokes to be missed. This can be done by one of three ways—by opening the switch in the ignition circuit, causing failure to ignite; by holding the exhaust valve open during the suction stroke; and by leaving the inlet valve closed during the suction stroke.

The actual thermal efficiency of an S.I. engine is $\dfrac{(\text{Work output})}{J(\text{Heat input})}$. The basis of output can be shaft power, or indicated (cylinder) power, while the input is generally taken as the *lower* heating value of the fuel. So if w_f lbs. of fuel were consumed in t time while the engine produced a measured power, P, the thermal efficiency is

$$\eta = \frac{P \times t}{J \times w_f \times \text{L.H.V.}} = \frac{2545t \times \text{HP.}}{w_f(\text{L.H.V.})}$$

in which HP. = Shaft horsepower and η = overall efficiency, or
HP. = Indicated horsepower and η = "indicated" efficiency.
t = Time, hours, taken to consume w_f.
w_f = Weight of fuel used, lbs.
L.H.V. = Lower heating value of that fuel.

An average heat balance for this type engine would be as follows:

Output	25%
Cooling	36%
Exhaust *	34%
Friction	5%
Total	100%

* Including incomplete combustion.

Another efficiency term is of special importance in the internal combustion engine, and no complete explanation of engine action is possible without invoking it. *Volumetric efficiency* may be defined as the weight of gas actually drawn in on an induction stroke, divided by the weight which would occupy the piston displacement under standard conditions of atmospheric pressure and 60° F. If an engine revolved very slowly, and the induction passages were large and unobstructed, the cylinder might be filled with a gas at practically atmospheric pressure, but still the volumetric efficiency could be less than 100% by the heating of this fresh charge through contact with warm manifold and cylinder walls. Since internal combustion engines rotate at speeds from 300 to 3000 rpm., a definite pressure decrement must be expected as necessary to overcome inertia and friction in order to get the cylinder filled with gas in so short an interval of time. Of course, the above refers to normal operation, as it is possible to obtain volumetric efficiencies higher than 100% by supercharging.

From the above it will be realized that volumetric efficiency has little in common with thermal efficiency, but depends on such factors as the rotative speed of the engine, the fraction of the cycle which is given over to induction, the shape of the ports and valves, and the temperature of the gas. The latter is affected by heating in manifolds, carburetor air heaters, or cylinders, although this may be partially offset by some refrigeration obtained in the vaporization action of the carburetor.

Example 1: A stationary S.I. engine constant-load test resulted in the following data. Brake horsepower 152, fuel consumption 55 lbs., elapsed time 40 min. Exhaust temperature 1250° F, cooling water flow 160 lbs. per min., in at 120° F, out at 175° F. How was the heat in the fuel distributed?

It will be assumed that a 15:1 air fuel ratio was obtained in carburetion, and that the average specific heat of the products is 0.28 B.t.u. per lb. per deg. The B.t.u. per min. accounted for in output, exhaust gas, and cooling can be calculated and subtracted from the heat in the fuel, giving as the difference, the combined friction, radiation, and incomplete combustion loss.

Lower heating value of fuel, $\frac{55}{40} \times 19{,}500 = 26{,}800$ B.t.u. per min.

Sensible heat in $\frac{55}{40} \times (15 + 1)$ lbs. exhaust $= 1.375 \times 16 \times .28(1250 - 60)$

$= 7380$ B.t.u. per min.

Cooling loss = $160 \times (175\text{–}120) = 8800$ B.t.u. per min.

Useful output = $152 \times \frac{2545}{60} = 6450$ B.t.u. per min.

Other losses = $26{,}800 - (7380 + 8800 + 6450) = 4170$ B.t.u.

HEAT BALANCE SUMMARIZED

	B.t.u./ min.	Per Cent
Output	6,450	24
Exhaust	7,380	28
Cooling	8,800	33
Friction, Radiation and Incomplete Combustion	4,170	15
	26,800	100

Example 2: A single cylinder engine 4 in. × 6 in. × 400 rpm., two-cycle, operates on the methane-air mixture whose heating value was computed on page 195. Using an indicated thermal efficiency of 20% and a volumetric efficiency of 62%, estimate the indicated horsepower.

$$\text{Displacement per minute} = \frac{4^2\pi}{4} \times 6 \times 400 = 30{,}200 \text{ cu. in.}$$

$$\text{Heat input per minute at 100\% volumetric efficiency} = \frac{30{,}200}{1728} \times 85.3 = 1490 \text{ B.t.u.}$$

Input at 62% volumetric efficiency = $1490 \times .62 = 924$ B.t.u. per min.

$$\text{Indicated horsepower} = \frac{924 \times 60 \times .20}{2545} = 4.36 \text{ hp.}$$

8–16. Supercharger. The performance of an internal combustion engine is indicated, among other things, by the brake horsepower output. A review of the factors affecting power indicates that atmospheric conditions have a significant effect. A naturally aspirated (unsupercharged) engine is able to draw into the cylinders on suction strokes only from 70 to 85% of the fuel charge which it is theoretically capable of inducing. Consequently, the mean effective pressures are not as large as they might be, and power output per cubic inch of piston displacement has not reached its maximum possible value. This volumetric efficiency is even lower at altitudes above sea level because air density is less. Compression of the incoming air, or air-fuel mixture, somewhat above ambient pressure is a natural way of increasing output at sea level or of regaining it at altitudes. A compressor used for this purpose is designated a *supercharger*.

The reader should be cognizant of the fact that raising the sea-level power of an existing engine by adding a supercharger is not ordinarily feasible, as the original design may be insufficient to withstand the increased structural stresses created by higher cylinder pressures. However, supercharging for the purpose of regaining sea-level rating at altitudes may be readily added

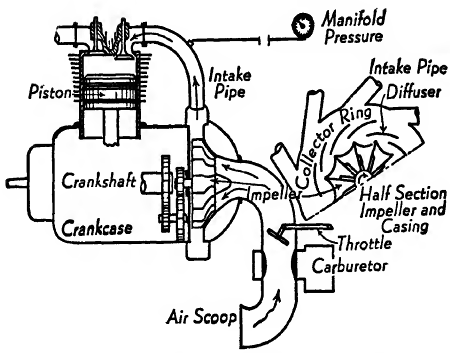


FIG. 8–28. Single-stage supercharger. Diagram emphasizes induction system.

provided the supercharger is not used at sea level and is employed only to offset the effect of altitude. Thus it is seen that supercharging falls into two categories: (1) supercharging that contemplates increasing capacity per cubic inch of piston displacement under approximately constant atmospheric conditions; and (2) supercharging that will keep power of an engine up near its rated output at high altitudes. The first category is sometimes termed "ground boosting," and the second "altitude supercharging." Obviously the latter predominates in the aeronautical field. The former will be found occasionally in stationary, marine, and surface vehicle usage of engines.

Either centrifugal compressors or positive displacement blowers can be used to supercharge. For ground boosting the latter are excellent, since boost varies linearly with speed, and so cylinder pressures and shaft torque hold up well under slow-speed operation, whereas the pressure boost given by a centrifugal compressor drops off rapidly as speed is cut. On the other hand, the

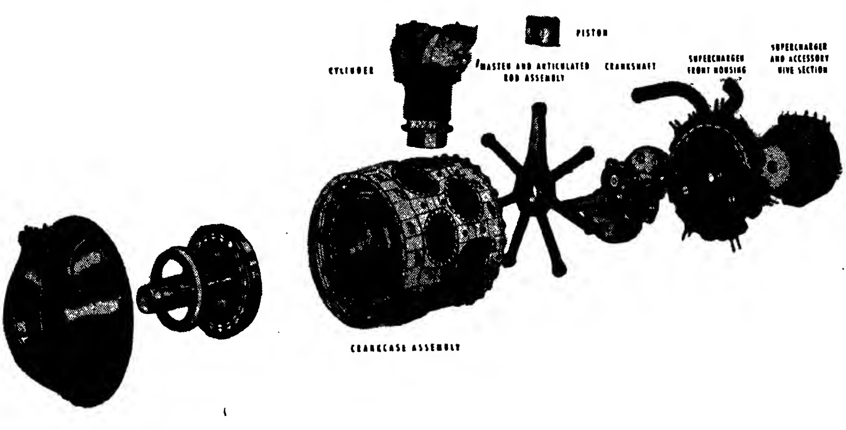


FIG. 8–29. Exploded view, two-row, 14-cylinder, radial engine. (Courtesy Wright Aeronautical Corp.)

displacement type suffers from excess bulk and weight compared to the centrifugal—a fact not likely to recommend it to the aeronautical field.

Sea-level S.I. engines are not often supercharged because they are most frequently found in small capacities where by relatively high rotative speeds engine size may be kept small. Furthermore, unless special high octane fuels are used, supercharging may create detonation troubles.*

High-altitude flights in airplanes would have been impossible without supercharging of the engine. Airplane engine superchargers are universally high-speed centrifugal types because of the emphasis on minimum size and weight. The principal types are: (1) the internal gear-driven type, receiving its drive from the crankshaft, and (2) the external exhaust gas turbine-driven type. Figure 8–28 shows the standard simple single-stage, single-speed, internal gear-driven supercharger, usually an integral part of large-capacity engines. The carburetor is located on the suction side of the impeller, so a

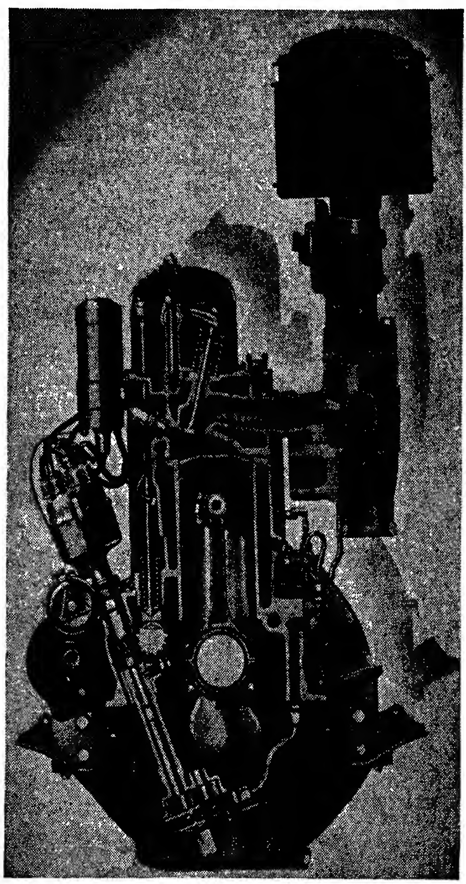


FIG. 8–30. Cross-section of an automobile engine. (Courtesy Chevrolet Motor Co.)

* Diesels, however, are frequently supercharged.

combustible mixture is being compressed. The duralumin impeller is geared up to make from 10,000 to 15,000 rpm., and is capable of boosting atmospheric pressure approximately one atmosphere. It would not be permitted to accomplish this at sea level, as the cylinder explosion pressures would be too high; also, detonation would probably occur. By throttling the intake at the carburetor, the cylinder pressures can be kept within safe limits. This is gaged by the pilot by the *manifold pressure;* that is, the mixture pressure just before it enters the cylinder. As an airplane climbs (engine rpm. constant) it is necessary to open the throttle gradually in order to keep manifold pressure constant. When the throttle is full open a *critical altitude* is reached, and further gain in altitude will be attended by decrease of output since there is no more supercharger reserve left.

PROBLEMS

1. Draw a p–V diagram of an ideal four-stroke Otto cycle, compression ratio 5, compression and expansion adiabatic, suction pressure 14 psi., maximum pressure 350 psi. Scales 1 in. = 20% piston displacement, 1 in. = 100 psi.

2. What is the air standard efficiency of the Otto cycle which has a compression ratio of 6?

3. What percentage clearance would be required to make the Otto cycle air standard efficiency 55%?

4. An engine with bore of $3\frac{1}{2}$ in., stroke $4\frac{1}{4}$ in., has a compression ratio of 4. How much is its air standard efficiency changed by shaving $\frac{1}{16}$ in. off the cylinder head-block parting surface?

5. A $3\frac{1}{2}$-in. $\times$ $4\frac{1}{2}$-in. engine was designed for use with high octane gasoline and a compression ratio of 7. At a time when low octane gasoline, only, is available, how much extra cylinder head gasket thickness must be inserted to reduce the ratio to 6.5?

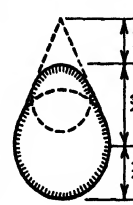
6. A four-cycle engine has rotative speed of 3000 rpm. Exhaust opens 40° before B.D.C., closes 5° after T.D.C. How much time, sec., is allotted to exhaust?

7. How long (sec.) is the inlet valve open on an engine running at 2500 rpm. if it opens 5° before T.D.C. and closes 30° after B.D.C.?

8. The timing of a certain four-cycle engine is described as follows: Exhaust opens 35° before B.D.C., exhaust through 220°, suction through 200°, and compression through 150° of crankpin travel. Overlap 10°. Draw a timing diagram similar to Figure 8–3.

9. The exhaust valve of an engine is open from 40° before B.D.C. to 6° after T.D.C. Angular advance of ignition 15°, valve overlap 11°, compression through 155°, all angles in terms of crankpin position. Draw a timing diagram.

10. Draw a diagram, similar to Figure 8–4, for the inlet valve having timing as follows: Opens 3° before T.D.C., closes 20° after B.D.C. Clearance .020 in., gear diameters 3 in. and 6 in. (full scale). Position of mechanism to be for beginning of suction.



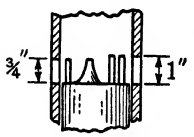
11. Same as Problem 10 except position is beginning of compression.

12. A poppet valve operating cam has a shape as specified in the accompanying figure. The tappet ends in a $\frac{1}{4}$-in. diameter roller which bears against this cam. Through how many degrees must

this cam turn in order to take up a clearance of .020 in.? Show graphical solution, scale 1 in. = $\frac{1}{2}$ in.

13. Assume that the answer to Problem 12 is 20°. How long (sec.) is the valve open at 1500 rpm. engine speed?

14. The accompanying sketch shows part of the lower portion of a two-cycle engine cylinder. Piston is on B.D.C. How much time (sec.) is allowed in this engine for the operation of charging the cylinder? Stroke 4 in., speed 1500 rpm., R/C 3.



15. Assume that the engine, for which a sketch accompanies Problem 14, has $2\frac{1}{2}$-in. crank, R/C ratio of 3, operates at 1200 rpm. How much time (sec.) is allowed for exhaust?

16. Same data as Problem 14. What fraction of the time of a complete cycle is given over to compression and power?

17. Same data as Problem 15. What fraction of the time of a complete cycle is allotted to scavenging?

18. The mixture actually drawn into an engine each cycle would only fill $\frac{3}{4}$ of the displacement under standard conditions. Fuel is gasoline, lean mixture 18:1, engine is 6 cylinder, four-cycle 3 in. $\times$ 4 in. $\times$ 2000 rpm. Find (a) the heating value per cubic foot of mixture at standard conditions; (b) the horsepower if the engine converts 15% of the heating value into output.

19. What is the ideal air-fuel ratio for alcohol CH_4O? What is the heating value per cubic foot of an ideal mixture at standard conditions? H.H.V. 10,720 B.t.u. per lb.

20. A quantity of gasoline-air mixture containing 90.6 B.t.u. weighs .0743 lb. After being compressed to 100 psi. the temperature was 340° F. It was then ignited and exploded at constant volume. Assuming $c_v = .20$, what theoretical final temperature do the products have? They actually are at 4200° F and 580 psi. Mention the reasons why actual conditions at the end of explosion differ from those calculated from $Q = wc_v\Delta T$ and Charles' Law.

21. What is the ideal air-fuel ratio, by weight and by volume, also the heating value of a cubic foot of mixture if the fuel is a by-product gas consisting 25% of CO and 75% N_2 by volume? Standard pressure and temperature.

22. Five gallons of iso octane are blended with 3 gals. of heptane. What is the octane rating of the blended fuel?

23. Assume that you are instructed to blend a gasoline of 90 octane with another of 70 octane and produce 5 gals. of 78 octane fuel. How much of each of the original fuels should you use?

24. In a simple Venturi metering carburetor the coefficients of discharge at a certain rate of flow are $C_a = .91$, $C_g = .62$. Air density at the throat is .07 lb. per cu. ft., gasoline density 55 lbs. per cu. ft. What does the gasoline jet diameter need to be for a 15:1 air-fuel ratio from a carburetor with a 1-in. diameter Venturi throat?

25. Assume that the answer to Problem 24 is .06 in., then work out the air-fuel ratio at an increased rate of flow through the Venturi when $C_a = .92$ and $C_g = .83$.

26. Assemble the component parts from illustrations furnished in this chapter, and diagram a carburetor of the following specifications. Principle — auxiliary air valve. Supply — float chamber. Auxiliaries — idling jet, choker. (Updraft, with butterfly valve.)

27. Same as Problem 26, with following change of specifications. Principle — plain tube. Supply — diaphragm chamber. Auxiliaries — idling jet and accelerating jet. (Updraft, with butterfly valve.)

28. Same as Problem 26, with following change of specifications. Principle — compensating jet. Supply — float chamber. Auxiliaries — idling jet, choker. (Updraft, with butterfly valve.)

29. Obtain a spark plug and (1) diagram it in section, to double its original size, (2) diagram the gap design, (3) record the manufacturer or his designation of the plug. Also label fully.

30. Make a full page ($8\frac{1}{2}$ in. $\times$ 11 in.) schematic diagram of a coil and battery ignition system for a four-cylinder automobile engine. Label fully.

31. Same as Problem 30, except for an eight-cylinder engine.

32. A 90-hp. water-cooled engine is estimated to have a thermal efficiency of 24% at full load. The cooling loss is 30% of the heating value of the fuel. If the "radiator" cools the water to 145° and 16 gals. are made to circulate per minute by the water pump, what is the engine outlet water temperature?

33. A V-type airplane engine is cooled with a 94% solution of ethylene glycol (which has a specific heat of 0.67 B.t.u. per lb. per deg. F, and specific gravity of almost 1) in water. The cooling jackets hold 7 gals., the radiator and piping 8 gals. Coolant out of engine 250° F, in at 160°. Engine is rated at 1100 hp. Assume 25% thermal efficiency, 28% cooling loss. Estimate the rate of circulation of the coolant — circuits per minute.

34. A large gasoline engine is cooled by the direct evaporation system. Power — 150 hp., assumed efficiency 26%, and cooling loss 29%. A steady stream of make-up water at 65° F is fed into the cooling system. What is the necessary rate of flow at full load?

35. Find the indicated horsepower of a six-cylinder engine, $3\frac{3}{4}$ in. $\times$ $4\frac{1}{4}$ in. $\times$ 2800 rpm. Four-cycle, single acting. Mean effective pressure 120 psi.

36. The engine pictured in Figure 8–1 has the following specifications: Brake hp. 31 at 2200 rpm. Bore $3\frac{1}{2}$ in., stroke 4 in. Calculate the brake m.e.p.

37. Engine test data: Fuel consumption .55 lb. per hp. hr., 19,000 B.t.u. per lb. Air-fuel ratio 16:1, exhaust temperature 1125°. Cooling water 1.25 lb. per min. per hp., heated 40° F. Compute a four component heat balance, assuming c_p of exhaust gas .28.

38. During an S.I. engine test the following data were obtained or deduced. Exhaust gas 15 lbs. per lb. fuel, temperature 1300° F, atmospheric temperature 75° F. Speed 3200 rpm. Fuel consumption 43 lbs. per hr., 19,100 B.t.u. per lb. Cooling water temperatures 180° F and 130° F, flow 10.5 gals. per min. Dynamometer torque 135 lbs. ft. net. Calculate a four-way heat balance and express in per cent.

CHAPTER 9

Compression Ignition Engines

9–1. Compression Ignition. Engines which can polytropically compress a gas to such an extent that it attains a temperature sufficiently high to initiate combustion may dispense with any other aids to ignition. Engines employing compression ignition are so different from those using electric sparks that the method of ignition becomes a worthwhile basis of internal combustion engine classification. As was pointed out on page 22, if $n > 1$ temperature rises during a polytropic compression. We may estimate the final temperature if (1) the compression ratio is known, (2) if it is assumed that the quantity of gas in the cylinder remains the same (i.e., no leakage), and (3) if the particular kind of polytropic process were known or assumed. The equation * required in this temperature study is $T_2/T_1 = (V_1/V_2)^{n-1}$.

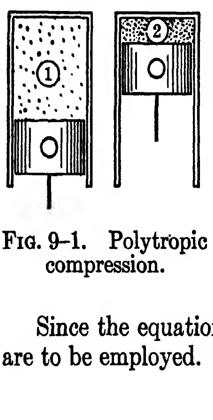


FIG. 9–1. Polytropic compression.

Example: State ① of the gas (air) diagrammed in Figure 9–1 is 14.7 psi., 60° F, with the piston on crank end dead center. State ② is at head end dead center. The compression ratio (V_1/V_2) is given as 15, and compression is adiabatic, i.e., $n = \gamma$. Assume $\gamma = 1.4$. The final compression temperature is to be estimated.

Since the equation cited above is founded on the gas laws, absolute temperatures are to be employed. So $T_1 = 460 + 60 = 520°$ R, $\gamma - 1 = 0.4$.

$$\frac{T_2}{520} = 15^{.4}, \text{ whence } T_2 = 1535° \text{R}.$$

Compression temperature = 1075° F.

Actual compression temperatures may fall somewhat below such computed values for these reasons:

1. The effective compression ratio is less than the ratio obtained from cylinder dimensions because, in addition to the possibility of a slight leakage of air past the piston rings, the cylinder ports are usually not completely closed when the piston is on crank end dead center.
2. Adiabatic compression is impossible on account of the need for cooling the cylinder of an I.C. engine. Hence some numerical value must be

* Such equations are easily obtained from the polytropic family equation and the general gas law.

assumed for n. It will exceed 1, but actual measurements show that it is generally somewhat less than γ, indicating that some heat is transferred during compression.

3. The compression is not of air alone. Each new cylinder charge is diluted with the residual products of the preceding cycle, which remained in the clearance space. Fortunately, the effect of this is practically negligible.

The Diesel cycle is a thermodynamic power cycle for I.C. engines. It is predicated on compression ignition, and is the foremost, and practically the only cycle in the compression ignition field. "Diesel" and "compression ignition" have almost become synonomous terms. The compression ratios of Diesel engines range from 12 to as high as 20. In any case the final compression temperature comfortably exceeds the ignition requirement of fuel oil.

The semi-Diesel engine is not a true compression ignition engine because the compression ratio employed will not compress the air sufficiently to cause ignition in a cold engine. The semi-Diesel engine, however, does not use electrical ignition in the manner of the gasoline engine. It has a hot-bulb, a mass of metal incorporated in the cylinder head in such a way that a portion of it projects slightly in the combustion space. Before starting, the hot-bulb is thoroughly heated by applying a blow torch to its exterior surface. It thus provides a focal center of high temperature which produces ignition during the starting of this type of engine. A similar service is performed by "glow plugs" in engines so equipped. The glow plug resembles a common spark plug except that the insulated electrode instead of terminating in a gap connects to a loop of resistance wire. When a moderate voltage is impressed on the plug the resistance wire glows and provides a focal high-temperature point.

9–2. Diesel Cycle. In a patent dated 1892, Dr. Rudolf Diesel, a German engineer, described an engine to operate on the Carnot cycle. Coal dust was the fuel, and it was to be fed rapidly enough so that isothermal expansion would result. After fuel cut-off, an adiabatic expansion would continue, followed by a compression made isothermal by the injection of water into the cylinder. An adiabatic compression then brought the cycle back to its beginning. A further claim of the patent covered the use of liquid fuels and the spray valve. Early attempts to build this engine resulted in the adoption of a modified cycle which, after much experimentation, was built into a successful working engine. Since then the Diesel has slowly but surely established for itself a secure position as an industrial and automotive prime mover.

Although modified from the inventor's original conception, the modern Diesel cycle retains the most important feature, namely that of compression of air to the ignition temperature, followed by timed introduction of fuel.

This cycle is shown in the accompanying diagram. The solid line indicates a theoretical cycle, the dotted line shows how a slow-speed actual cycle may depart from the theoretical. Beginning with point a on the cycle, imagine that a cylinder filled with air is closed at the end by a tightly fitting piston. The piston is moved to compress the air without addition or loss of heat through the cylinder walls. As the air is decreased in volume, the pressure rises adiabatically, and it arrives at the condition corresponding to point b. As the piston starts to move, continuing the cycle from point b, fuel is injected into the cylinder just rapidly enough so that its combustion will maintain the pressure while the volume is being increased, at least up to point c. At c when the outward stroke is partially completed, the fuel is *cut off*, and the products of combustion expand adiabatically from c to d, giving work to the piston as they do. At d the exhaust valve opens, and the pressure drops to a. The line extending horizontally from a represents the theoretical exhaust and suction stroke. Adiabatic expansion and compression are not possible in a cylinder which must be well cooled in order to maintain a lubricating oil film. Therefore an actual cycle will not be expected to follow the adiabatic. By assuming adiabatic compression and expansion, only, in the cylinder, and no mechanical friction, an expression may be derived for the efficiency of the cycle *abcd*. This, the so-called *Air Standard Efficiency*, is given by the following equation:

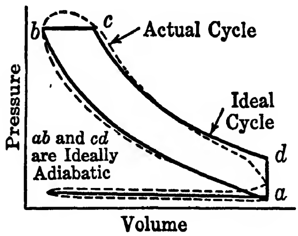


FIG. 9–2. Diesel cycle.

$$\eta = 1 - \frac{1}{\gamma r^{\gamma-1}}\left[\frac{R^{\gamma} - 1}{R - 1}\right],$$

in which r = Compression ratio.

R = Cut-off ratio, V_c/V_b.

γ = Adiabatic polytropic exponent.

On account of the idealistic assumptions on which the air standard efficiency is based, its use produces values much in excess of actual engine performance. It is, however, informative of the effect of r and R upon Diesel efficiency, and sets a goal which engine designers may struggle to achieve. Note that to improve capacity (represented by the area of the cycle) one must sacrifice efficiency, for prolonging the process bc increases R and diminishes η.

Example: What is the ratio, which might be called *internal efficiency*, between the actual and air standard efficiencies of a Diesel whose clearance is 7% and which operates with a fuel cut-off at 10% of the stroke? Measured thermal efficiency is 35%.

The clearance is used to obtain both r and R.

$$r = \frac{100 + 7}{7} = 15.28; \quad R = \frac{7 + 10}{7} = 2.43.$$

$$\eta = 1 - \frac{1}{1.4 \times 15.28^{.4}} \left[\frac{2.43^{1.4} - 1}{2.43 - 1}\right] = 58.6\%.$$

$$\text{Internal efficiency} * = \frac{35}{58.6} = 59.7\%.$$

9-3. Mixed Cycle. High-speed internal combustion engines perform their thermodynamic cycles of operation so rapidly that, whether originally intended to operate on the Diesel or the Otto principle, the actual cycle has a combustion that exhibits both constant volume and constant pressure phases. Spark ignition engines must receive the timed spark in advance of piston dead center position. Compression ignition engines must have the injector timed to begin the fuel spray also ahead of dead center position. In either case, on account of high rotative engine speed the combustion is only partially completed before the piston begins the expansion stroke. The remaining fuel is burned approximately at constant pressure. A conventionalized cycle is shown in the accompanying figure. Combustion extends from b to d. The air standard efficiency expression is:

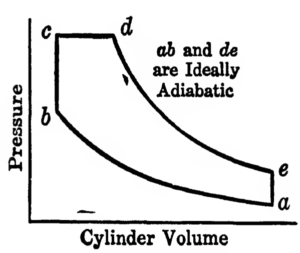


FIG. 9-3. Mixed (or combination) cycle.

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \left[\frac{ZR^{\gamma} - 1}{(Z - 1) + \gamma Z(R - 1)}\right],$$

in which Z = Combustion pressure ratio P_c/P_b.
$R = V_d/V_c$.
r and γ as in the preceding section.

All factors which influence either Otto or Diesel cycle efficiency appear here, and in addition a factor introducing characteristics of the constant volume phase of combustion.

9-4. Diesel Engine. The Diesel, being an I.C. engine, has much in common with spark ignition types. Because the latter have previously been described, no detailed examination is included here of C.I. engine features that are similar to those of S.I. engines.

* Of the cycle. Not to be confused with mechanical efficiency.

The two types have the following in common:

1. Mechanically they follow the same general pattern.
2. Both are constructed in two- and four-cycle types.
3. They are similarly cooled and lubricated.
4. The valve gear action is similar.
5. As high-speed engines, both operate on the "mixed" cycle.
6. Both require an external starting system.

They differ as follows:

1. The compression stroke acts on an air-fuel mixture in the S.I. engine—on air in the C.I. engine.
2. The source of ignition is basically different.
3. S.I. engines are restricted to gaseous or easily vaporized fuels.
4. Four-cycle C.I. engines always have "I" head cylinders.
5. C.I. engines usually have replaceable cylinder liners, whereas only heavy-duty S.I. engines are given this treatment.
6. C.I. engines have no electric ignition system, no carburetor, a simple inlet manifold, but require a fuel injection system. S.I. engines have a simpler fuel supply system.
7. The ideal Diesel cycle is founded on isobaric, the ideal Otto on isometric, combustion.
8. S.I. types are found in a greater variety of cylinder arrangements, C.I. types over a greater range of physical size.
9. Control of most S.I. engines is by varying the quantity of an air-fuel mixture of constant quality, whereas C.I. engines are controlled by varying the fuel input to a constant air charge.

The Diesel engine aspirates an air charge, or has it rammed into the cylinder, depending on whether it is four- or two-cycle. This air is then compressed enough to raise its temperature to 800°–1000° F. Just before the end of the compression stroke the fuel system will begin to spray a finely atomized fuel oil into the hot air. The spray continues for several degrees of crankpin travel past dead center. The injected fuel burns violently, liberating heat, raising the temperature of the products, and maintaining the pressure against a tendency to decrease as volume expands. When the fuel spray is cut off the pressure does begin to diminish rapidly, but throughout the remainder of the power stroke gas pressure works expansively on the piston. So, a considerable quantity of heat energy is transferred to mechanical work *à la* Scheme No. 1 of Chapter 4. When finally the cylinder ports are opened or uncovered near the end of the stroke, the gas pressures and temperatures are quite moderate.

As was mentioned, aviation engines exhibit the greatest variety of cylinder arrangements. It has not been possible to construct compression ignition

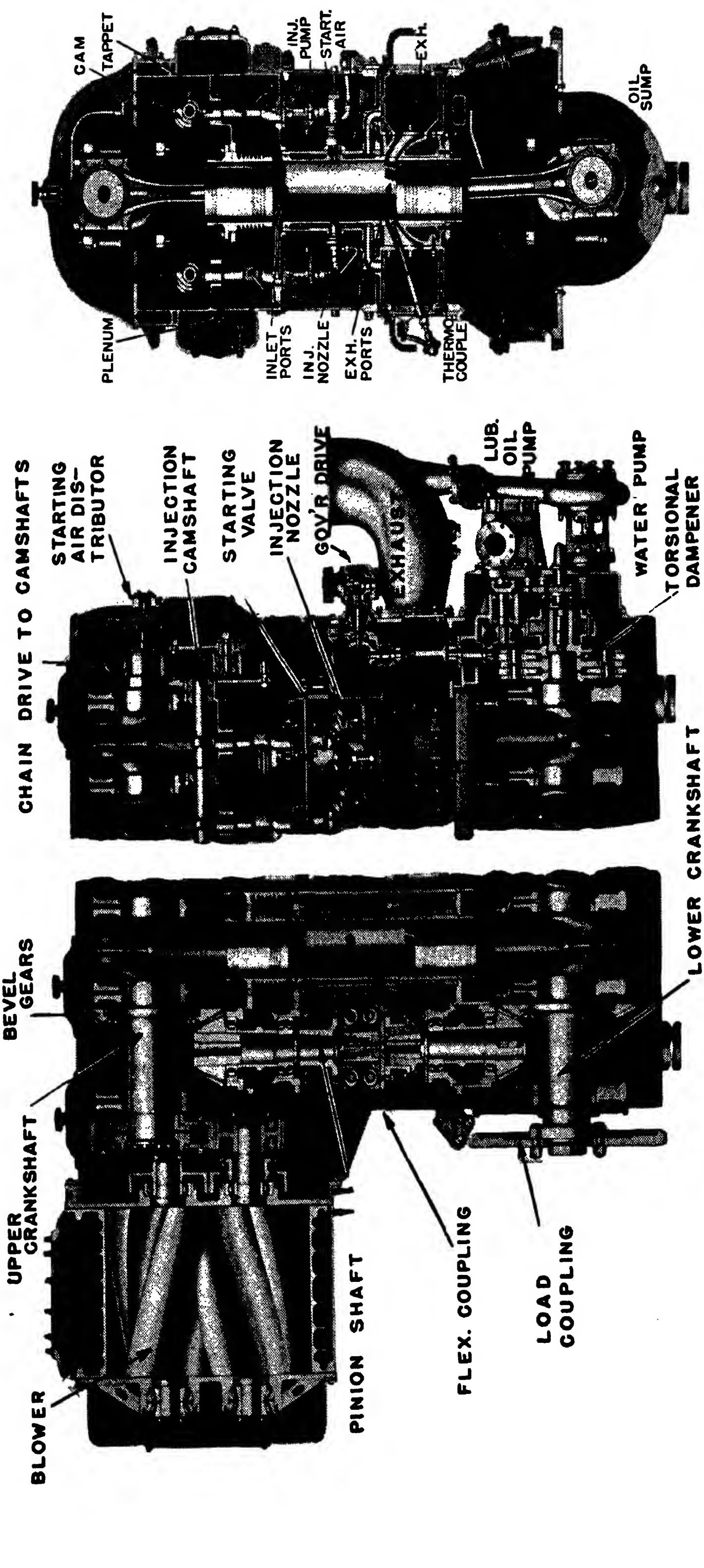


FIG. 9–4. Opposed piston engine. Long- and cross-sections. (Courtesy Fairbanks Morse & Co.)

engines with weight-per-horsepower ratios as favorable as those yielded by spark ignition designs. Other fields of application are covered by relatively few variations of cylinder arrangement. Diesel engines are usually built with upright, in-line cylinders, sometimes en bloc (automotive practice), sometimes with separate cylinders on a common crankcase (common stationary practice). A few designs will be found in V-bank arrangement, and some in the opposed piston design, an example of which is shown by Figure 9–4. These special types have been developed for applications requiring an engine of high capacity but of compact design. Their rotative speeds and comparative weight are intermediate between the slow-speed stationary types and the high-speed automotive Diesels. Typical uses include railway locomotives, tug boats, and submarines.

Most stationary Diesels are of the two-cycle type, and most automotive Diesels are four-cycle. Stationary Diesels are relatively slow speed. As air, only, is inducted, the problem of pre-compression is simpler than in S.I. two-cycle engines. Automotive Diesels are more difficult to build because much stress is laid on light weight and high rotative speed. Completing the cycle in four strokes instead of two seems to be of very material assistance in overcoming these difficulties; at least, the four-cycle type predominates in this field. Because they are materially different physically, the two- and four-cycle types will be treated separately, but in any event the Diesel engine must possess:

1. The slider crank chain elements.
2. Valves and valve gear. The piston is its own valve in two-cycle designs.
3. Fuel injection system.
4. Lubrication system.
5. Cooling system.
6. A means of control.
7. Starter.

9–5. Two-Cycle Diesel Engine. The true Diesel cycle will be more exactly reproduced in an engine if plenty of time is allotted for the completion of a cycle. Slow speed is also suited to two-cycle design. Hence one important type of these engines is a slow-speed two-cycle design. Adequately charging the cylinder with fresh air for each cycle is one of the principal considerations —and a point of considerable variation among the different manufacturers. The air which blows the exhaust gas out of the cylinder and becomes the working fluid for the next cycle is called *scavenging air*. This is furnished by crankcase compression, as in Figure 9–5, or by blower, as in Figure 9–4. Scavenging air can be admitted, and exhaust released through ports located where the piston will uncover them just before bottom dead center is reached. However, in some cases these lower cylinder ports are all for air inlet, and

mechanically operated valves in the cylinder head open to release the exhaust gas.

The engine shown in Figure 9–5 is as simple as any internal combustion engine can ever be. Having crankcase compression, no blower is required. It is a slow-speed (360 rpm.) design, and satisfactory scavenging is obtained without auxiliary valves. Being of the "open chamber" design the cylinder

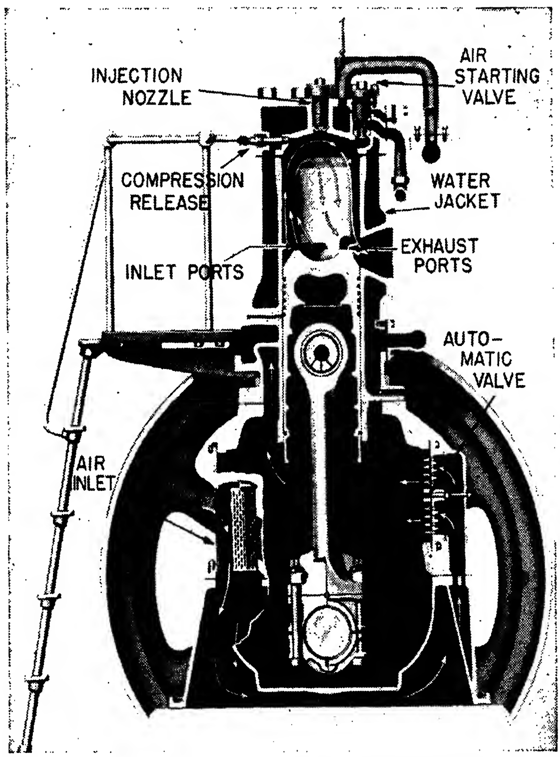


Fig. 9–5. Two-cycle stationary type Diesel engine. (Courtesy Fairbanks Morse & Co.)

and combustion chamber are of simple shape. There is no cylinder liner, contrary to usual Diesel practice, but an extra long piston minimizes cylinder wear. Such engines are rugged, reliable, and efficient. They are bulky for their capacity, and have a high specific weight, i.e., pounds per horsepower. In stationary power plant prime movers this usually is no disadvantage. The external appearance of this engine is shown in Figure 9–6. A two-cylinder engine is shown; however, engines of various sizes can be assembled, using the same basic equipment, except crankcase and crankshaft, but with different number of cylinders.

Describing the operation of this engine, the cycle may be said to begin with the closing of the exhaust ports by the piston on the compression or up-

stroke, approximately 50 degrees past the bottom dead center, when the compression of the charge of fresh air begins.

Compression continues to within a few degrees of top dead center when injection occurs and spontaneous ignition takes place due to the high temperature of the compressed air. After combustion, during which the piston passes its top dead center position, the piston continues its accelerated downward movement on the working stroke due to expansion of the gases.

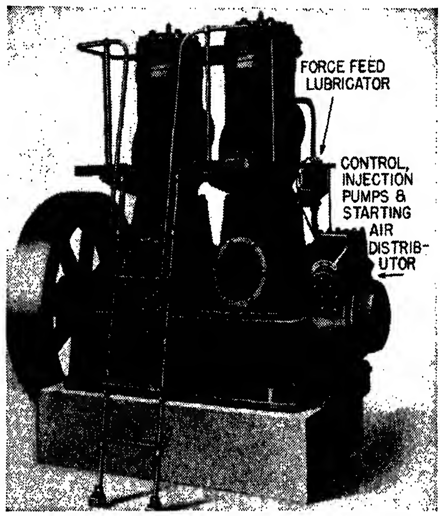


FIG. 9–6. External appearance of the engine diagrammed in Figure 9–5.

When the piston has reached a point about 50 degrees ahead of bottom center, the exhaust ports are uncovered and the burnt gases at a pressure slightly above atmospheric are released to the exhaust system.

As the piston moves farther down, the scavenging air ports are uncovered and air under slight pressure is admitted to the cylinder from the crankcase. The entering air is directed upward along the cylinder wall on the side opposite the exhaust ports by the shape of the air passages and the contour of the piston head and thence drives the remaining burnt gases out the exhaust ports on the other side of the cylinder.

On the next upstroke, first the air ports and then the exhaust ports are covered and compression begins.

The fuel injection valve and nozzle is located in the center of the cylinder head and discharges directly into the cylinder. A few degrees before compression has been completed, near the top dead center of the piston travel, an accurately metered quantity of fuel is introduced into the cylinder through the multi-orifice nozzle. The charge is ignited by the heat of compression, and combustion takes place at a pressure of about 650 lbs. per sq. in.

On the upstroke of the piston, air is drawn through a screen and automatic

air valve into the crankcase and is there compressed on the working or down stroke of the piston to a pressure of a few pounds. The exhaust ports open ahead of the air ports, thus reducing the pressure within the cylinder to atmospheric. Then, when the piston uncovers the air ports at the end of the down

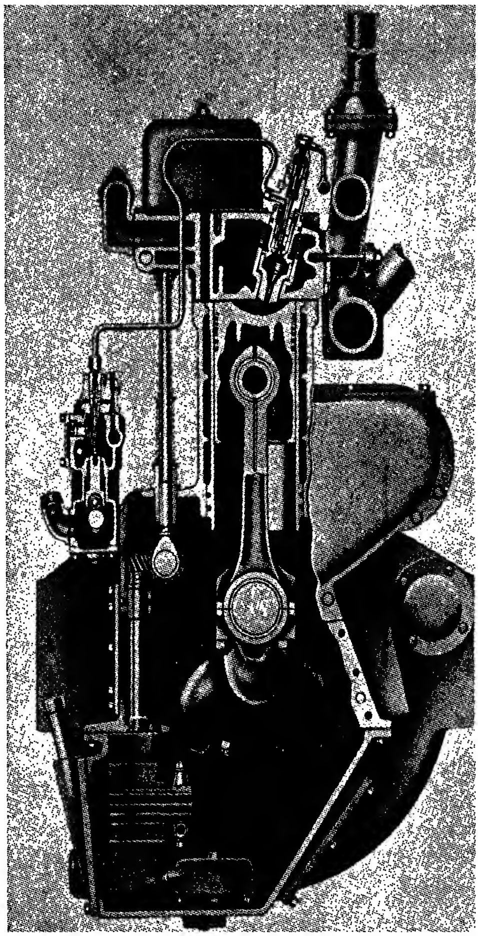


FIG. 9–7. Cross-section of Diesel engine. (Courtesy Caterpillar Tractor Co.)

stroke, this air rushes into the cylinder and effects scavenging of the burnt gases.

Passages for conducting the scavenging air from the crankcase to the cylinder are so arranged that the air enters the cylinder directed toward the wall opposite the exhaust ports. As indicated in Figure 9–5, this air follows the backwall, so-called, and makes a complete traverse of the cylinder up one side and down the other, driving the exhaust gas ahead of it.

9–6. Four-Cycle Diesel Engine. In the early period of Diesel engine building the four-stroke cycle was commonly employed with the large slow-

moving stationary engines which formed the backbone of Diesel sales and development. Gradually the problems of the two-stroke cycle were overcome, and a definite trend to that cycle set in. Later, as builders sought to compete with the S.I. engine in the automotive field, they found it necessary

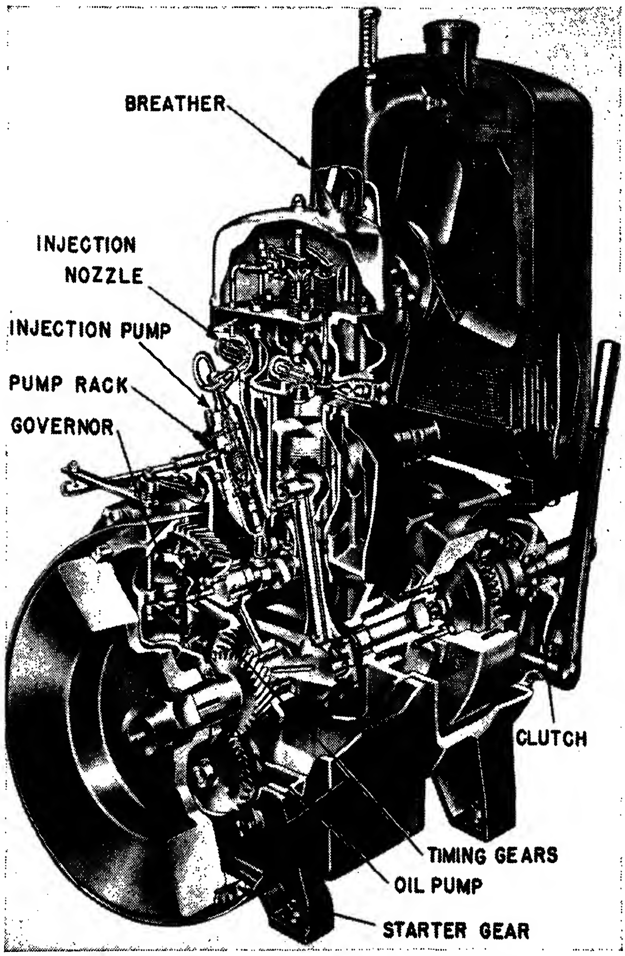


FIG. 9-8. High-speed C.I. engine. (Courtesy Atlas Imperial Diesel Engine Co.)

to specialize in the four-cycle principle because (1) high rotative speeds were required in order to reduce size and weight per horsepower, and (2) four-cycle action is superior in efficiency at high speeds. Also, the four-cycle engines control better for variable speed and power. More adequate time for the actions of scavenging and injection are allowed by the four-stroke principle. Volumetric efficiency is higher, starting is easier. The majority of automotive Diesel engine builders use the four-cycle principle in spite of the fewer power strokes and the additional mechanical parts of the valve gear. To obtain

good clean combustion in the short period allotted at customary automotive speeds, these designs have special combustion chambers to promote gasification of the fuel oil and turbulence of the burning fuel-air mixture. Slow-speed, two-cycle action obtains this turbulence without special treatment of the combustion chambers, which are therefore usually simple in design. Combustion chambers will be considered at greater length in a later section.

The single-cylinder portable power unit shown in Figure 9–8 is a high-speed, four-cycle compression ignition engine. The engine has a $3\frac{1}{8}$-in. stroke, $3\frac{3}{4}$-in. bore, and compression ratio of 15. Normal speed under governor control is 1800 rpm. The cooling medium is water, and it can be seen that the removable cylinder sleeve is of the wet liner type. The aluminum alloy piston has three compression and two oil control rings. Valves are overhead to secure the small clearance volume required for compression ignition. They are operated by rocker arms which in turn are pushed by rods from tappets actuated by cams located on the camshaft in the crankcase. The combustion chamber embodies the Lanova energy cell principle, which is described in Section 9–7. Injection equipment consisting of pump and spray nozzle is joined by a short thick-walled delivery tube. The pump is direct cam-driven from the camshaft. A centrifugal governor is also camshaft mounted. The engine has full pressure lubrication, pressure being supplied by a rotary, gear-driven oil pump on the crankcase base. A large cooling radiator and water connections are used because circulation of the cooling medium is accomplished by thermo-syphon action. Although not shown, this engine has an electrical system consisting of a 12-volt, V-belt-driven generator, storage battery, ammeter, starting switch, and starting motor. Also needed are an air filter, lubricating oil filter, fuel oil transfer pump and filter. Fuel for an engine of this type should possess a Baumé value of 30 to 40, and a cetane number between 46 and 60.

9–7. Combustion. Comparing the conditions of combustion in C.I. and S.I. engines, as is done graphically in Figure 9–9, oil is sprayed into the highly compressed and dense air for compression ignition, but the fuel and air are already thoroughly mixed in the case of spark ignition. It is apparent that whereas approximately ideal air-fuel ratios are suitable (as well as necessary) in S.I. engines, there needs to be considerable excess air present in the C.I. engine combustion chamber because there is no pre-mixing of fuel and air, and because the fine droplets of oil have a hard time penetrating the compressed air. The slow-speed stationary Diesels, although efficient, are quite conservatively rated as to maximum temperatures and mean effective pres-

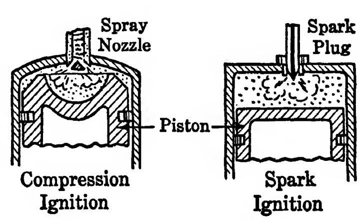


FIG. 9–9. Combustion chamber comparison.

sures because cut-off ratios R are low and the air-fuel ratio at full load may be as much as twice the theoretical. At part load the ratio will be even higher because C.I. engines induct the same quantity of air per cycle at all loads—but vary the quantity of fuel injected into it. On the other hand, with the assistance of special aids to gasification and turbulence, the automotive Diesel operates with a full load air-fuel ratio of 18 or 20:1, representing very little excess air. The nature of compression ignition combustion being isobaric, and the air usually being in some excess as just mentioned, temperatures reached by the products are not conducive to much thermal dissociation.

Example 1: With reference to Figure 9–2, and using the same nomenclature, let it be required to analyze isobaric combustion in terms of cut-off ratio, maximum temperature, and air-fuel ratio. Given the ratio 25:1, lower heating value of fuel oil 18,500 B.t.u. per lb., clearance 7%, temperature at state b 1000° F. Heat transferred to the cooling system during combustion is neglected.

The constant pressure combustion from b to c during which 25 + 1 lbs. mixture receive 18,500 B.t.u., represents a rise of temperature ΔT, where $\Delta T = \dfrac{18{,}500}{26 \times c_p}$. Assuming $c_p = .295$, $\Delta T = 2410°$. Then $T_c = 3870°$ abs. Since $T_b = 1460°$ abs., $V_c/V_b = \frac{3870}{1460} = 2.65$. This is R. To find the percentage of stroke during which injection occurs, let x = that percentage.

$$R = \frac{7 + x}{7} = 2.65; \quad x = 11.62\%.$$

Example 2: With further reference to the same engine, what air-fuel ratio and injection are allowable, assuming complete combustion, if the temperature is permitted to rise to 3600° F. Clearance 7%, t_b 1000° F.

$$R = \frac{3600 + 460}{1000 + 460} = 2.78.$$

$$R = \frac{7 + x}{7}; \quad x = 12.5\%.$$

$$w = \frac{Q}{c_p \Delta T} = \frac{18{,}500}{.295 \times (3600 - 1000)} = 24 \text{ lbs.}$$

Therefore the air-fuel ratio is 23:1.

The fuel ordinarily employed is refined crude petroleum. The low cost of good-grade petroleum fuel in the United States has, to the present time, precluded the general use of any other type of Diesel fuel. Fuel oil is the residue left when distillation has removed the gasoline, kerosene, and light distillates from the crude. The heavier residues are also removed from the better grades of fuel oil. Comparable qualities of fuel oil were mentioned on pages 56–57. Like the S.I. engine, the C.I. engine suffers from the use of improper fuels. The most important adverse effect derived from fuel is detonation. Pecul-

iarly, detonation is aggravated in the C.I. engine by that very quality which suppresses it in the S.I. type. A slow-burning fuel is anti-knocking with spark ignition, a fast-burning one with compression ignition. To discover the reason for this, remember that fuel oil must be sprayed into hot air, mixed, and ignited. Both the mixing process and the rise of temperature of the fuel are time-consuming, and an *ignition delay* occurs which must be offset by advancing the moment of fuel ignition before dead center position. The number of degrees of crankpin travel between beginning of injection and the action of ignition is termed the delay angle. It is desired to have the delay angles as small as possible so that the cylinder will not be overloaded with fuel when ignition occurs.

It is believed that combustion knock in a Diesel engine is caused by the delay in ignition during the first part of the injection, causing accumulation of fuel which burns simultaneously with the fuel injected during the latter part of the injection. Good *ignition quality*, as represented by high Cetane Number, minimizes the delay period and the tendency of the engine to knock. This is a characteristic which measures the ability of the fuel oil to ignite spontaneously in the engine cylinder. It is a quality which assumes increasing importance the higher the rotative speed of the engine.

The *Cetane Number* scale is derived from the standard practice of using a test fuel composed of cetane and alpha-metholnaphthalene. Of these two, cetane has very good ignition qualities, while alpha-metholnaphthalene is very poor. The Cetane Number of a fuel is the percentage of cetane, in a mixture of the two, which will give the same ignition quality as the fuel oil under test. The average high-speed Diesel engine requires a fuel having ignition quality of better than 45 Cetane Number, and will scarcely run at all on fuels rated lower than 25 Cetane. Cracked oils show a considerably lower Cetane Number than straight run oils.

Diesel engine combustion chambers exhibit great differences. To understand why, note first that a fuel jet of good atomization will have poor penetration, and consequently poor mixing with the air. The same would be true of a solid jet which, while possessing good penetration power, would expose little surface and fail to obtain mixing. In a slow-speed engine a fuel jet of moderate atomization and good penetration will be granted sufficient time for complete combustion without detonation in a plain or "open" combustion chamber. In high-speed engines, however, one sees many variations in combustion chamber design. These might be classified as *turbulence, precombustion,* and *air cell* chambers. Such cylinder heads have auxiliary chambers or pockets incorporated for the purpose of promoting a high degree of turbulence. The fuel spray is received in both the turbulence and precombustion chamber but incomplete combustion predominates in the latter since approximately a third of the air in the combustion chamber is driven into the precombustion

chamber, whereas over three-quarters of it is compressed into the turbulence chamber. However, in either case the fuel is sprayed into the auxiliary chamber, whereas the main chamber receives the fuel in an air cell type. The air cells usually contain even less of the compressed air than a precombustion chamber will. As soon as piston travel lowers combustion chamber pressure, the air cell contents are ejected into the main chamber, creating turbulence.

The *energy cell* is a variation of the air cell, in which fuel is intentionally sprayed into the cell. The Lanova system (diagrammed) consists of a combustion chamber and parts so designed as to create a great turbulence of fuel and air in the chamber. The objective of extreme turbulence is a better mixture of air and fuel to produce complete and even burning of fuel.

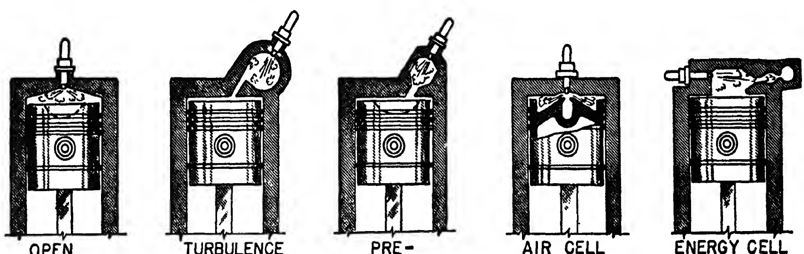


FIG. 9–10. Types of combustion chambers.

Fuel passes from the fuel nozzle into the main combustion chamber where it is partially mixed with hot compressed air generated by the compression stroke. A portion of this mixture passes through a Venturi orifice into what are termed minor and major energy cells where a very rapid combustion takes place. This discharges back into the main combustion chamber where it impinges on such residual spray as is still leaving the nozzle and is in suspension. This backfire, or counterflow, produces a violent turbulence, giving progressive rise in pressure by accelerating the end of the combustion cycle instead of the beginning.

9–8. Fuel System. The function of the fuel system is (1) to *pressurize*, (2) *to meter*, and (3) to *time* the input of fuel to the combustion chamber. The injection of fuel into a Diesel cylinder is a precise, split-second operation. A minute quantity of fuel oil must be injected as a spray into air compressed to 300 lbs. per sq. in. and over. It must become well mixed with the air and burn almost instantaneously, yet at a definite rate so that pressure rise can be controlled. In a 100 hp. cylinder of an engine operating at the moderate speed of 750 rpm. the amount of each injection is approximately two *drops* of oil. The injection must be started, completed, and the burning finished all in approximately .01 sec. This picture is illustrative of quantities for one of the larger Diesels, at full load. Part-load operation and smaller power

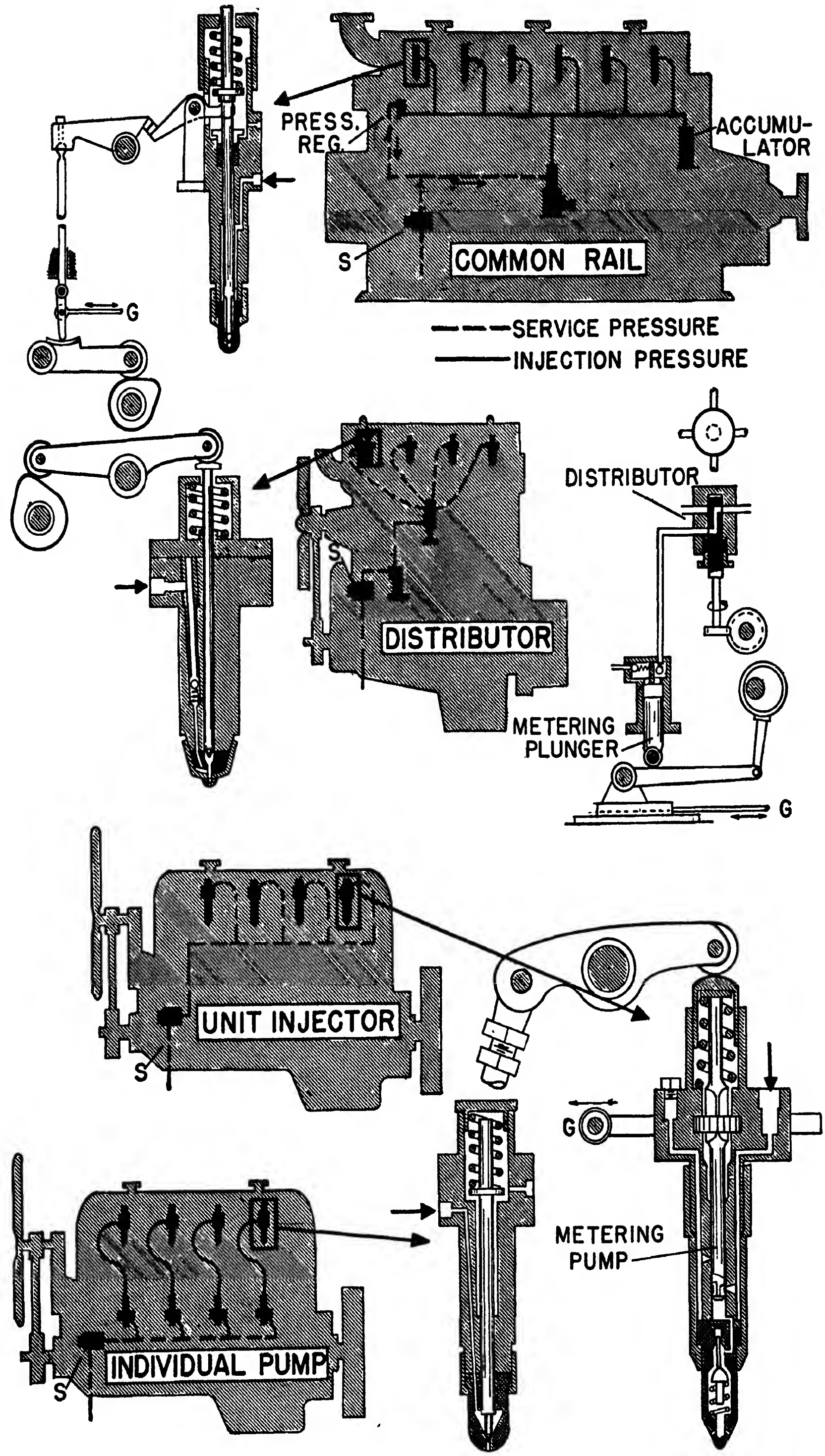


FIG. 9-11. Systems of fuel injection. Modern Diesel engine designs exhibit many combinations and/or modifications of these basic systems. For simplicity, drain lines, filters, and valves have been omitted. S—Fuel supply pump; G—Governor or throttle control.

capacities would each decrease the volume of fuel oil injected. To meter such a tiny volume accurately under the action of speed responsive governors and to inject it with proper timing and atomization into a region of "hard" air (i.e., highly compressed) requires the very best of machine work in construction of the injection pumps (which are the heart of the system) and of the injection valves.

The pumps are small diameter plunger type pumps which usually either open a by-pass for cut-off of fuel or regulate fuel suction rather than curtail the plunger stroke. The arrangements of injection on multi-cylinder engines could be classified as follows:

Individual Pump System. In this, the most popular system, an injection pump is provided for each engine cylinder. A spring-loaded injection valve is provided in each cylinder and the pump discharge is connected to it by high-pressure capillary-type tubing.

Common Rail System. A single pump maintains injection pressure in a common header from which branches lead to the injection valves. But now, the pumps not providing metering of fuel, the injection valve must be mechanically operated (cams and rocker arms, etc.) to provide the timing and metering function.

Distributor System. This may also be a single-pump system, but with the injection valve drive relieved of the need for metering the fuel. A single injection pump meters the fuel for all cylinders—at low pressure. The metered fuel is delivered through a rotating distributor valve which shunts the metered fuel to the various injection valves. The mechanically operated valve plungers then raise the oil pressure sufficiently for spraying. Thus the valve action times and pressurizes the fuel but does not meter it.

Unit Injector System. In this system the functions of the pump and injection valve are combined in a single mechanical unit attached to the cylinder in the usual location of the injection valve. It must be mechanically operated and the method of actuation must include the timing, metering, and pressurizing function. Oil is furnished to the unit injectors from a common header supplied by a low-pressure pump.

In the early development of Diesels the best results were secured with air injection wherein small quantities of highly compressed air were employed at the injection valve to drive the fuel into the combustion chamber in a finely atomized state. The necessity of a high-pressure air compressor (usually three-stage with intercoolers) was a disadvantage, so when builders were able to overcome the problems of airless, or "solid" injection the air injection principle was gradually abandoned.

Figure 9–13 shows a differential *injection valve* suitable for use in an individual pump system. It contains a spring-loaded needle valve which is a lapped fit in a barrel or guide which is in turn fitted into the main valve body.

Through drilled passages in the body and passages formed by external fluting on the bushing the fuel reaches the valve tip, the passage through which is closed by the needle valve. When the pressure at this point reaches a predetermined value the needle is lifted off its seat against spring tension and fuel is forced through the passage in the tip and into the cylinder through multiple orifices. At the end of the injection pump stroke the fluid pressure

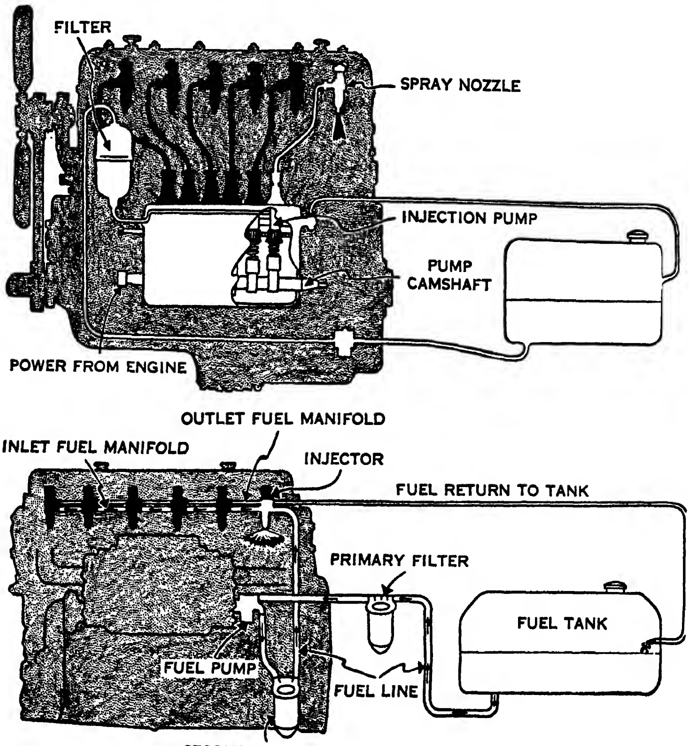


FIG. 9–12. Location of the elements of an automotive type fuel system. (Courtesy General Motors Corp.)

immediately drops and the needle valve reseats itself, thus cutting off the flow of fuel to the cylinder. The multiple orifices of the tip are so drilled and are of such size as to secure the most advantageous combination of turbulence and penetration to effect complete combustion.

How a Diesel engine *injection pump* operates is shown in Figure 9–14. The pump there shown would not be suitable for an actual installation, as it does not have any means of varying the quantity pumped per stroke. The metering function will be introduced later. Since a discharge pressure of 2000 psi. and up is not an uncommon requirement, and whereas the quantity pumped is ordinarily minute, a very small bore plunger pump of short stroke is in order. For positive timing and a powerful spray at the very outset of injec-

tion, as well as a quick tight cut-off, only an intermediate part of the plunger stroke should be used. During the first and last portions of the stroke, the cam is moving the plunger slowly so those portions are discarded. It will be seen that the head of the plunger is grooved and that two ports are located in the pump cylinder wall. When the plunger is on B.D.C. the cylinder fills with fuel from the supply pump. As the plunger rises it drives fuel out of the

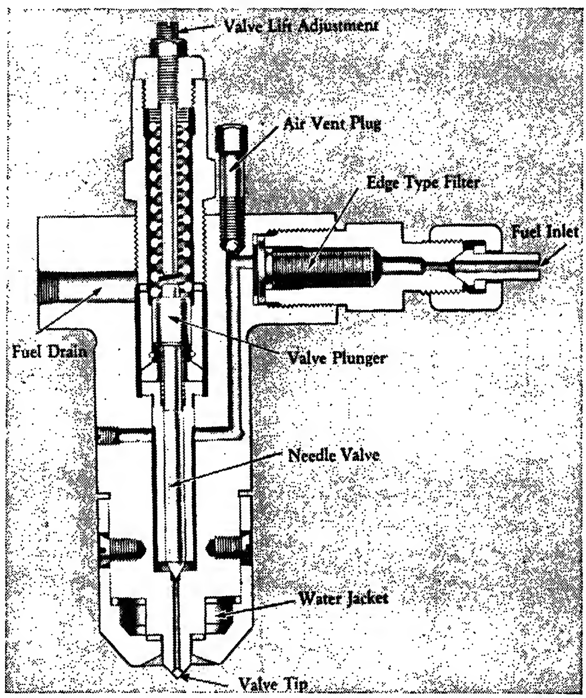


FIG. 9–13. Spring-loaded injection nozzle. (Courtesy Fairbanks Morse & Co.)

ports until it has risen to cover them. Then almost instantly the pressure increases sufficiently to force the fuel through the discharge line to the injection nozzles and thence into the combustion chamber. Delivery, which begins upon coverage of the ports by the plunger, ends abruptly when the relief groove registers on the by-pass port. The pressure is quickly dissipated to the by-pass and the injection nozzle valve snaps shut. However, the plunger completes its full stroke L before starting to return. The portion of the stroke during which fuel was pumped into the cylinder is designated the *effective stroke*, L_e.

We will now see how with a slight modification this pump can be made to meter the fuel quantity by varying the effective stroke. In Figure 9–15 the plunger pump has had the groove enlarged so that an edge of it has become a helix. Also the cam follower has teeth on its rim which engage a straight gear

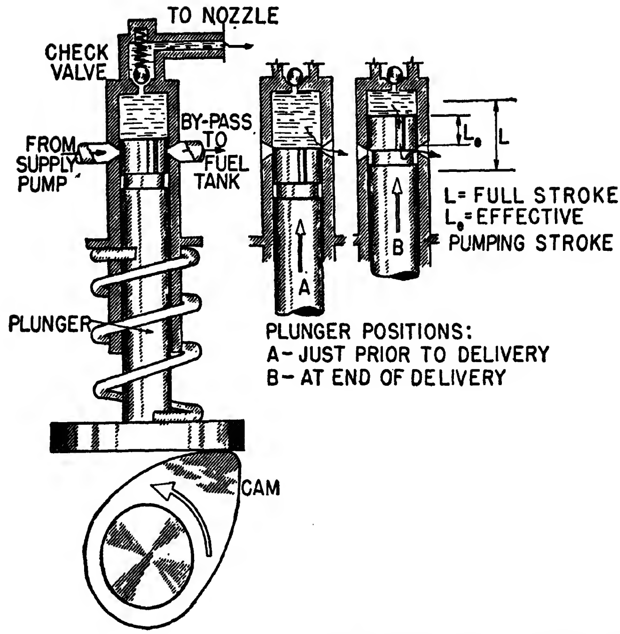


FIG. 9-14. Principle of the plunger type injection pump constant load.

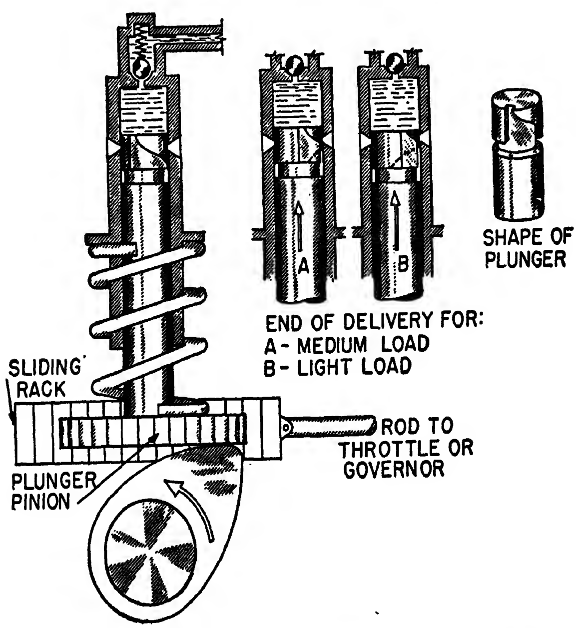


FIG. 9-15. Principle of the plunger type injection pump variable load.

rack whose height is enough to keep the teeth in mesh even though the pinion reciprocates. By moving this rack horizontally, the plunger can be rotated a little, even while it is reciprocating. Study of the figures should show that the effect of the helical cut is to vary the plunger position at which the relief groove registers with the by-pass port. Since delivery begins at the same point, but terminates differently, depending on the rack position, movement of the rack can cause a change in effective stroke and therefore, also in fuel pumped per stroke and power developed by the engine. Actual metering pumps may differ from this considerably in mechanical detail, though adhering to the principle.

While the injection pump metering system just described is in common use, there are competing systems such as variable plunger stroke, controlled suction, and separately driven by-pass valves.

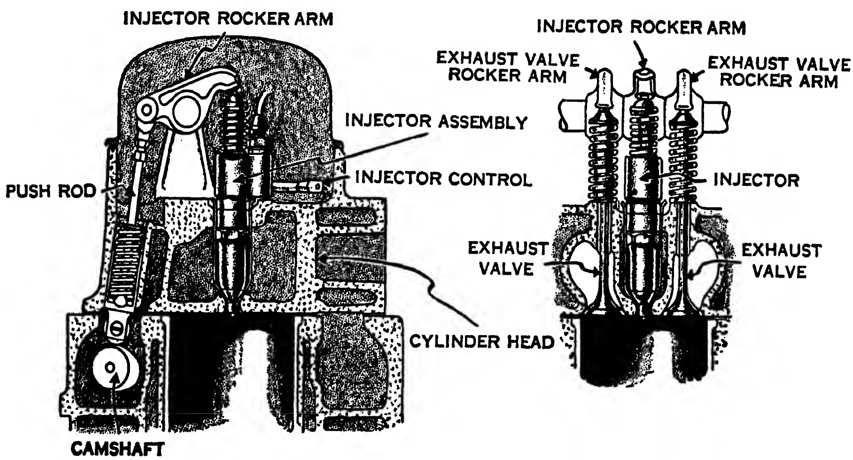


FIG. 9-16. Unit injector in a two-cycle engine with exhaust valves. (Courtesy General Motors Corp.)

Compression ignition engines, with their closely fitting fuel pump plungers, and their injection nozzles with tiny orifices, require fuels that have no suspended solids. They are much more sensitive to fuel condition than are the carburetted engines, since the carburetor jet openings of the latter are comparatively large. Fuel injection engines are featured by the care taken to pass the fuel through high-class filters before injection. In all systems a surplus of fuel is fed by the supply pump to the high-pressure pumps in order to maintain a small pressure head on the pump intake. The surplus drains back to the supply tank or reservoir.

9-9. Cooling and Lubrication. Practically all C.I. engines are water cooled. Automotive types generally follow S.I. engine practice in cooling and lubrication. Stationary units exhibit a greater variety of cooling treatment. Only in rare instances can the cooling water be wasted after it has passed

through the engine. Even on marine engines modern practice is to recirculate fresh water through the engine and cool it with raw water in a surface heat exchanger. Engines cooled directly with salt water must be specially modified for resisting the action of electrolysis and salt incrustation. Water must be let out cooler, creating a greater cooling loss than in fresh water cooling engines.

Cooling systems are pictured in Figures 9–17 and 9–18. Stationary Diesel engines are frequently cooled with water that is (1) circulated through a cooling tower or (2) cooled by a raw water circuit which includes a cooling tower.

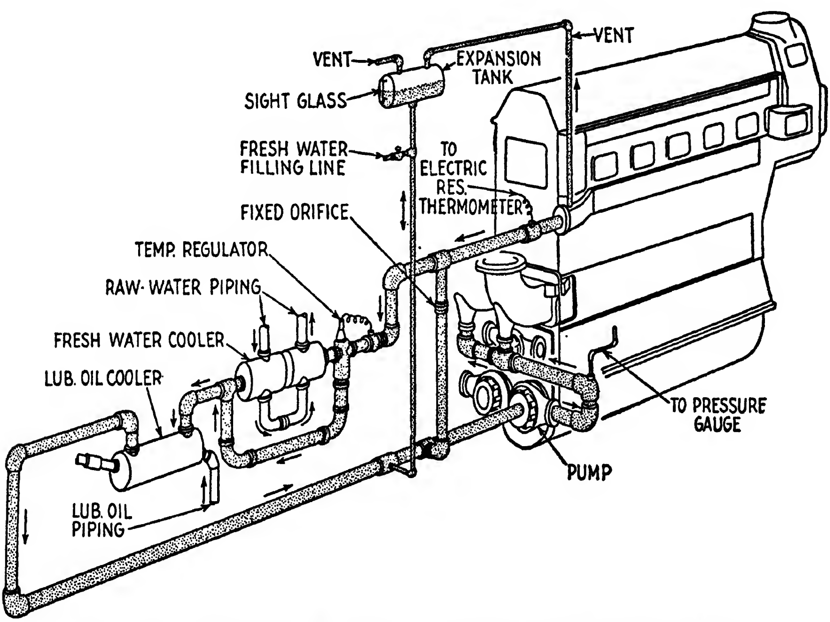


Fig. 9–17. Closed cooling system. (Courtesy Fairbanks Morse & Co.)

Quantity of water needed for cooling may be ascertained by the same method as used in Chapter 8. In assuming heat quantities, remember that the Diesel is more efficient, that the per cent cooling loss is about the same, and that the fuel is of slightly lower heating value.

Good lubrication is absolutely essential to this type of engine. Engine life is very dependent on this. Rarely does a machine possess the stringent lubrication demands of the C.I. engine and few exhibit as varied practice in lubrication. The principal systems are:

1. Mechanical force feed throughout. A lubricator unit consists of multiple pumps, one for each point lubricated. Pumps are individual because in the case of a common pump, if one feed line became partly clogged, all the oil would pass to the other points. Engines so lubricated show a large amount of small lubricator tubing externally.

2. Continuous pressure circulation. A pump delivers oil under pressure to all necessary points by means of tubes or passages cast in the crankcase. Distribution to connecting rods and piston follows standard automobile practice.

3. Splash. The oil level in the crankcase is carried high enough for the connecting rods to dip in it and splash oil throughout the interior of the engine. Generally some parts, such as cylinder walls, must be reached by force feed lubrication.

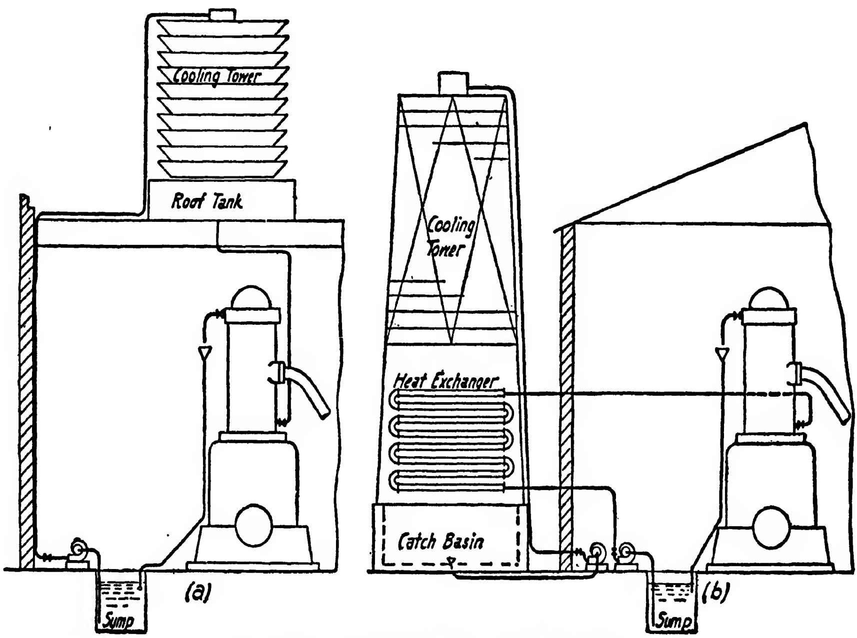


FIG. 9–18. Diesel engine cooling systems.

Filtration and centrifuging are employed to clean sediment from lubricating oil in order to be able to recirculate it. Cooling also is frequently needed.

9–10. Capacity and Performance. By capacity we generally mean horsepower rating. Compression ignition engines are built with capacities of from several thousand horsepower per engine down to as small as 5 hp. Most automotive types are more than 30 and less than 100 hp., and few of any other type exceed 1500 hp. per engine. Capacity is allied to brake mean effective pressure (b.m.e.p.) and rotative speed. The *brake m.e.p.* is that imaginary pressure which, if acting on the piston of a frictionless engine, would produce the actual brake horsepower. Since this kind of power is yielded by the formula * HP. = $PLAn/33{,}000$,

$$p_{\text{bmep}} = 33{,}000 \times \text{Brake HP.}/LAn \text{ lbs. per sq. in.}\dagger$$

* See page 125. † L in ft., A in sq. in.

Automotive engines must be as light and compact as possible consistent with a reasonable degree of stamina. They use higher b.m.e.p.'s, and higher rotative speeds than other C.I. engines.

Typical performance of compression ignition engines is displayed in Figure 9–19. Graph A is typical of a slow-speed stationary engine, B a high-speed truck engine. Both achieve about the same minimum specific fuel consumption, but the high-speed engine must be furnished with a superior quality, and therefore more expensive fuel. The slow-speed engine test curves

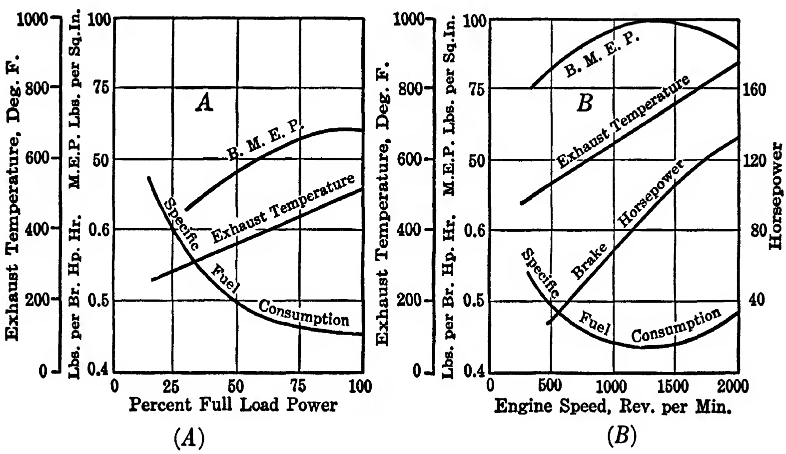


FIG. 9–19. Typical diesel engine performance.

A. Slow-speed two-cycle, constant speed.
B. High-speed four-cycle.

represent constant speed operation. The other data were obtained by loading the engine at different speeds until limiting ignition conditions were indicated by smoky exhaust, high exhaust temperature, excessive b.m.e.p., etc.

Comparing the efficiency of C.I. with S.I. engines of the same type—say truck engines—typical figures at the point of most efficient operation could be 0.097 gal. gasoline per brake hp. hr. for the S.I., and 0.064 gal. fuel oil per brake hp. hr. for the C.I. engine. To obtain a clear picture of relative efficiency these figures should be modified by considering the density of the two fuels and the heating values. In addition, the difference in unit fuel costs should be considered in comparing economy of operation.

Example: Data on fuels for the two engines just mentioned are:
Gasoline. 6.2 lbs. per gallon, 18¢ per gallon, 19,500 B.t.u. per lb.
Fuel oil. 7.2 lbs. per gallon, 10¢ per gallon, 18,500 B.t.u. per lb.
Compare the performance by (a) thermal efficiency, (b) cost of a hp. hr.

(a)

$$\eta_{\text{S.I.}} = \frac{\text{output}}{\text{input}} = \frac{2545}{.097 \times 6.2 \times 19{,}500} = 21.7\%.$$

Similarly,

$$\eta_{\text{C.I.}} = \frac{2545}{.064 \times 7.2 \times 18{,}500} = 29.8\%.$$

(b)

$$\text{Fuel cost of an S.I. hp. hr.} = .097 \times 18 = 1.746¢.$$

$$\text{Fuel cost of a C.I. hp. hr.} = .064 \times 10 = 0.64¢.$$

From these calculations it is concluded that whereas the C.I. engine uses 66% as much volume of fuel as the S.I. does, its efficiency is 37.2% better and its cost of operation only 37% as much.

Lest it be thought that such comparative figures should spell the end of S.I. engines, remember that the above are fuel costs only. When amortization of the purchase cost of the engine, and cost of repairs, are included, the overall picture is liable to favor S.I. types unless the expensive C.I. engine is used rather steadily (as it is in stationary generating plants, auto bus fleets, marine propulsion, etc.).

9–11. Starting C.I. Engines. Small engines follow S.I. practice. They have electric cranking motors for starting. The cranking torques are much greater, due to the higher compression ratios of C.I. engines, and 12- or 24-volt electric equipment is required instead of the 6-volt systems common to automobile engines. While hand cranking is a possible alternative method on gasoline engines, it definitely is not on Diesels. Large engines are started by admitting compressed air to the cylinders so that the initial motion is gained as in an air motor. Some manufacturers supply the compressed air to one cylinder, only, of the engine, and have the exhaust valves propped open on the other cylinders so as to reduce the torque required. Others supply compressed air to all cylinders at the time of the normal power impulse by means of a rotary distributor valve. The starting air comes from supply tanks which have to be recharged after each start. A small gasoline engine-driven compressor is therefore a common Diesel plant auxiliary.

Engines such as on large tractor, bulldozer, or portable power units, are sometimes out of the range for electric cranking, but a compressed air plant would be impractical to attach. These engines often have a small one- or two-cylinder gasoline engine as an integral auxiliary for starting. The gasoline engine is started by hand cranking, then mechanically clutched to the Diesel for the purpose of starting the latter. After the Diesel starts firing the starter engine is un-clutched and stopped.

In warm weather a Diesel in good condition will start more easily than a gasoline engine because the fuel reaches the cylinder on the first turn of the crankshaft, whereas several turns may be needed before carburetted gasoline appears in the cylinders of those engines. Conversely, in cold weather the gasoline engine can generally be started at once by the expedient of "choking" the carburetor inlet. But the Diesel compression will sometimes fail to

heat the air to ignition temperature because of excessive heat transfer to extra cold cylinder walls. Electric intake manifold heating and electric glow plugs in the combustion chambers are two methods sometimes used to assist in cold starting.

9–12. Supercharge. When more air is forced into the cylinder than would be there with a 100% volumetric efficiency, the condition is known as "supercharged." The reader is asked to distinguish between scavenging—which is

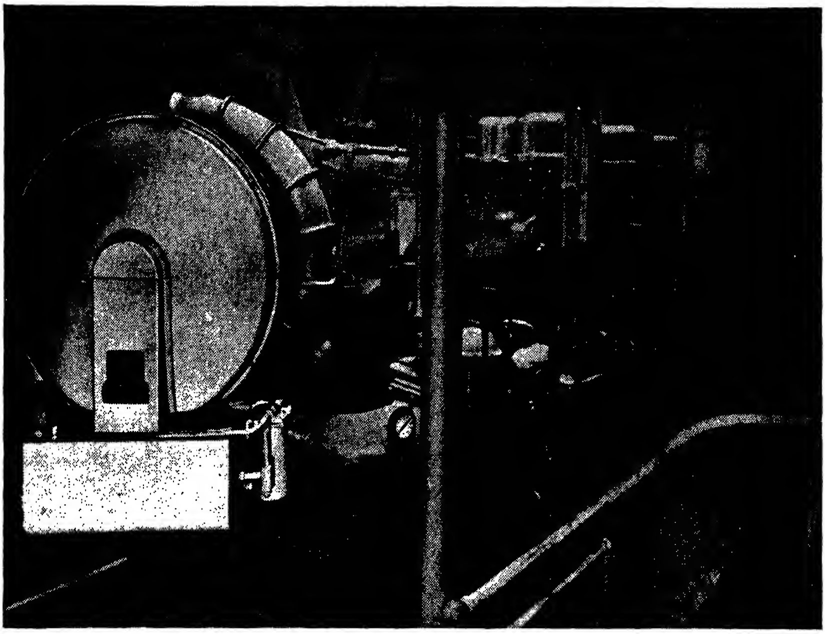


FIG. 9–20. Büchi system turbo-supercharger. (Courtesy Elliott Co.)

the use of plenum to assist in obtaining a normal induction of air on two-stroke designs—and supercharging, which consists of pressurizing the intake to the extent that a plenum exists in the cylinder at the beginning of compression. More air is present in the cylinder, and so more fuel can be burned without exceeding certain limiting temperatures. Brake m.e.p.'s are higher, and the same piston displacement is capable of an increased power capacity. But bearing loads are greater, cooling problems are accentuated, etc., so such engines must receive special design treatment. Diesel supercharging is usually resorted to only when space is at a premium, and maximum possible output per cubic inch of piston displacement is being sought. This occurs most frequently in marine and locomotive service.

Positive displacement blowers like that in Figure 9–4 are used both for scavenging and for supercharging. An excellent and efficient method is to supercharge with a centrifugal compressor driven by an exhaust gas turbine.

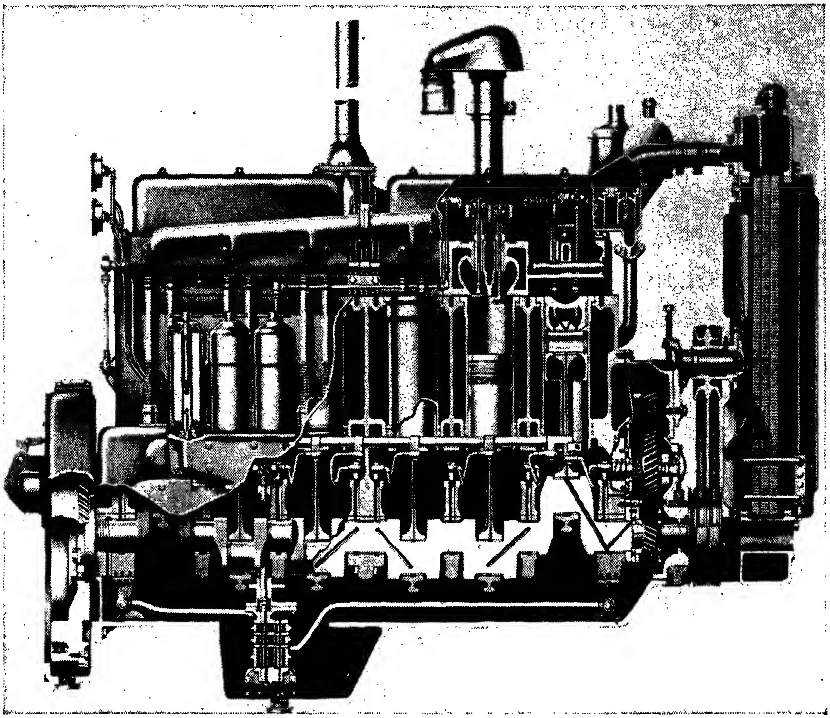


FIG. 9-21. Heavy duty mobile type unit (Courtesy Caterpillar Tractor Co.)

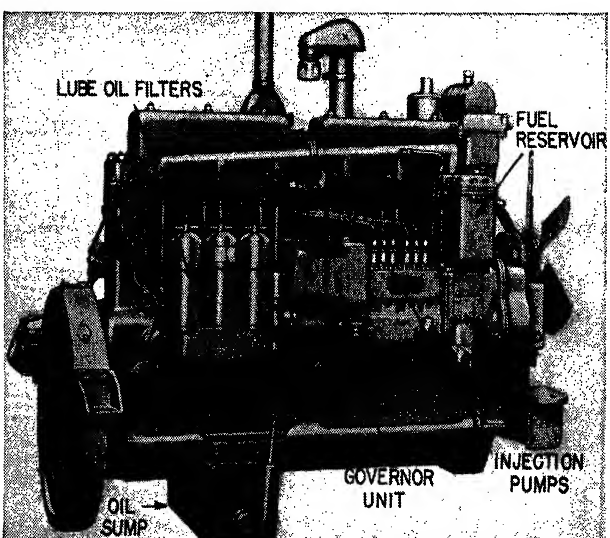


FIG. 9-22. Exterior of a heavy duty mobile engine. (Courtesy Caterpillar Tractor Co.)

The gage pressure existing in the gases at release from the engine is employed to expand them through turbine nozzles. Energy thus released is captured by a turbine wheel which in turn drives the compressor.

Figure 9–20 shows a turbo-charger installation on a 1200-hp. Diesel engine installed in a sea-going tug. The supercharging unit which is in the foreground receives exhaust gas from the engine manifold and, after expanding it, discharges it into a vertical exhaust stack.

The Buchi system of turbo-supercharging is peculiarly effective in obtaining good gas turbine performance without unduly raising the back pressure in the engine. This is accomplished by proper design of the exhaust manifold, especially its length, so that, resonant oscillation being set up, the gas turbine nozzles will receive a gas ram and an improved inlet pressure. Advantages also lie in overlapping the valve opening on four-cycle types so that some supercharging air can blow through the exhaust ports and cool the valves.

PROBLEMS

1. Find the theoretical compression temperature of air, given initial temperature 95° F and clearance 7%.

2. What compression ratio is required to achieve a theoretical 900° F compression temperature in air that was initially at 70° F?

3. As density is increased, the ignition temperature of a mixture of fuel oil and air is decreased. Tests show a relation between ignition temperature and density, represented by pressure, as follows: $t = (15{,}000/p) + 380$; t = ignition temperature deg. F, p = compression pressure, psi. abs. (for $50 < p < 500$). Plot this curve; also the polytropic compression curve for initial t of 80° F, and initial p of 15 psi., using $n = 1.35$. Shade the area between them and interpret the physical significance of its height, also the point of intersection. Scales 1 in. = 100 psi. abscissa, 1 in. = 100° F ordinate. Ranges 50–500 psi., 300° F – 1000° F.

4. A certain Diesel engine under test developed a maximum efficiency of 40%. The compression ratio is 16 and cut-off ratio 3. Using 1.35 in place of γ, find the percentage of the theoretical efficiency actually achieved.

5. The clearance of a Diesel cycle is 5%. Fuel is cut off after 10% of the stroke is completed. Find the air standard efficiency.

6. What is the air standard efficiency of a Diesel cycle having a compression ratio of 15 and a cut-off at 12% stroke?

7. Plot an ideal Diesel cycle for $r = 12$, $R = 1.835$. Initial state, p 15 psi., V 1 c.f. Scales 1 in. = .2 c.f., 1 in. = 100 psi.

8. A combination cycle has $r = 12$, $Z = 1.25$, $R = 3$. Initial state p 15 psi., V 1 c.f. Plot the ideal cycle to scales of 1 in. = .2 c.f., 1 in. = 100 psi.

9. What is the air standard efficiency of a mixed cycle having clearance of 10%? Pressure increase by explosion is 80 psi., cut-off at 10% of stroke.

10. Analyze whether an increase of Z (of the mixed cycle) will increase or decrease air standard efficiency.

11. An opposed piston engine with a bore of $4\frac{1}{2}$ in., crank radii 3 in., is to have a compression ratio of 15. How far are the piston faces apart at H.E.D.C.?

12. An engine similar to Figure 9–5, revolving at 360 rpm., has stroke of 10 in.,

bore 8 in. R/C 4. 6% of the stroke suffices to cover the by-pass ports, 10% to cover the exhaust port. How long (sec.) will scavenging air be blowing exhaust out of the cylinder?

13. An automotive C.I. engine operating on the mixed cycle is expected to have reliable ignition if the compression temperature is made to exceed the auto-ignition temperature by 200°. Compression is according to $PV^{1.3} = C$, from 14 psi. and 85° F. Employ the equation stated in Problem 3. Maximum pressure of the cycle is to be 400 psi. Find the percentage clearance, and Z. (Hint: Plot the equation mentioned against $T_2/T_1 = (P_2/P_1)^{(n-1/n)}$ and find p_2 for a ΔT of 200°.)

14. Assume that the answer to Problem 13 is a compression ratio of 10, then find temperature and pressure at the end of explosion, assuming no heat transferred during explosion.

15. Using the equation cited in Problem 3, find the clearance required to produce a compression temperature 250° in excess of the ignition temperature of fuel oil. Initial conditions 75° F and 14 psi. Allow for non-adiabatic nature of the compression by allowing $n = 1.35$ instead of γ.

16. A Diesel engine using 28 lbs. of air per lb. of fuel (18,000 B.t.u. lower heating value) has ignition occurring at H.E.D.C. at 800° F. Combustion is isobaric. Clearance 8%. Through what percentage of the power stroke will injection be required? $c_p = .295$. Neglect heat transfer to jackets during combustion.

17. Solve Problem 16, had 5% of the heat of the fuel been transferred to cooling during injection.

18. During combustion in a C.I. engine 18,000 B.t.u. are added to 25 lbs. mixture, initially at 850° F, while the displacement increases from 0% to 9%. Clearance is also 9%. During combustion 1000 B.t.u. enter the combustion chamber cooling system. Was this combustion approximately isobaric? Support your answer with a combustion calculation.

19. Find the percentage of stroke at fuel cut-off and the air-fuel ratio that should be used if the following conditions of isobaric combustion are met. Heat (net) added to products of 1 lb. of fuel, 17,500 B.t.u., complete combustion, c_p .295, clearance 10%, temperature at ignition 820° F, maximum temperature 3500° F.

20. Suppose we have a fuel injection pump with a plunger .3 in. in diameter. The maximum effective stroke is .65 in. What size cylinder (i.e., horsepower) would this serve if we assume the following? Four-cycle, 1200 rpm., 18,500 B.t.u. fuel, 55 lbs. per cu. ft., thermal efficiency 32%.

21. What are the weight and volume of the fuel required per cycle for a 2000 hp., 6 cylinder C.I. engine whose brake thermal efficiency is 34.6%? Oil is used 22° Baumé. Engine is two-cycle, 100 rpm. Estimate higher heating value with: H.H.V. = 17,680 + 60 × Baumé degrees. Use hydrogen composition of 15% in estimating L.H.V.

22. A single cylinder of a certain Diesel engine produces 80 hp. at full load, 300 power cycles per min. Thermal efficiency 35%, fuel oil 18,500 B.t.u. per lb., .90 specific gravity. (a) How much heat must be produced per power stroke? (b) What volume (cu. in.) of oil will possess this heat value? (c) What effective stroke of the oil pump plunger would inject this if plunger diameter is $\frac{1}{4}$ in.? (d) Does this answer imply a need for ordinary, excellent, or superlative machine work in building the pump?

23. The data of Figure 9–19B is for a 6-cylinder, 4-cycle engine whose stroke length is 30% more than its cylinder diameter. What is the bore x stroke of this engine?

24. What is the brake mean effective pressure of a four-cycle engine whose output is 60 hp. at 1700 rpm.? Bore $3\frac{1}{2}$ in., stroke $4\frac{1}{2}$ in. Engine displacement 260 cu. in.

25. Use the data of Figure 9–19A to estimate the sum of friction and cooling losses at full load. Assume air-fuel ratio of 28. Record any assumption of data found to be necessary.

26. Repeat Problem 25 for Figure 9–19B with air-fuel ratio of 18.

27. Using 30% as the cooling loss for the engine test shown in Figure 9–19B, estimate the quantity of cooling water required (gals. per min.). Water temperature rise in the engine is 50° F.

28. The volumetric efficiency of a supercharged two-cycle Diesel engine is 150%. Engine dimensions are: 6 cylinder, 5 in. × 8 in. × 750 rpm. With an assumed air-fuel ratio of 24:1, and thermal efficiency of 38%, what brake m.e.p. might be developed? Fuel 18,250 B.t.u. per lb.

29. Compute the air standard efficiency of the cycle described in Problem 7.

30. Compute the air standard efficiency of the cycle described in Problem 8.

CHAPTER 10

Gas Turbines

10–1. Historical. The gas turbine as a prime mover has only recently "arrived." Although in conception it is by no means new, few persons knew of it * or understood its possibilities prior to 1930. Many of those who had investigated the gas turbine concluded that it was impractical, or uneconomic. The metallurgical requirements of the gas turbine were so difficult to meet, and the S.I. and C.I. engines so well developed that experimentation with the gas turbine tended to be discouraging. Nevertheless, from 1900 on there was almost always some development work going on in this field. Many years were still to pass before slow progress could overcome the commercial defects of very low thermal efficiency, and small net power output available after satisfying the needs of the compressor.

Leonardo da Vinci's sketches (1550) included a crude chimney-gas-operated turbine, probably meant to be used to turn roasting spits over the fire. Later these rudimentary gas turbines were often built above medieval fireplaces, being called "smoke jacks."

In the nineteenth century several gas turbine systems were patented, and several were constructed in the early 1900's, but results were not auspicious. In some the efficiency was practically nil—that is, all the turbine output was used up by the compressor. Others of better efficiency were as complicated mechanically as the I.C. engines.

It became apparent that the upper temperature of the cycle must be very high, say 1200° to 1500° F, and the compressor and turbine be made exceptionally efficient if the gas turbine were to achieve a competitive position among other prime movers. Metallurgical progress, especially during the 1920–1930 decade, a better understanding of fluid mechanics, and valuable experience gained in developing the I.C. engine exhaust gas supercharger, have at last served to establish the gas turbine as a practical competitor of the internal combustion engine—at least under favorable conditions. Many successful installations have been completed since 1935, and the field was broadened to include mobile as well as stationary power.

10–2. Principle of Action. The reader will recall the comparison made between engine and turbine principles in Chapter 4. How power is produced

* Except in the exhaust gas supercharger form.

by engines using the products of combustion as the working medium was explained in Chapters 8 and 9. Similarly the gas turbine produces mechanical power, using products of combustion, *but* by the turbine principle which utilizes conversion of heat into mechanical kinetic energy of a jet, followed by the absorption of this mechanical energy by a rotating, bladed wheel.

In the gas turbine a stationary nozzle discharges a jet of gas (usually products of combustion) against the blades on the periphery of a turbine wheel, as shown in Figure 10–1. The jet is thereby deflected and slowed while the blades receive an impulse force which is transmitted as a mechanical torque

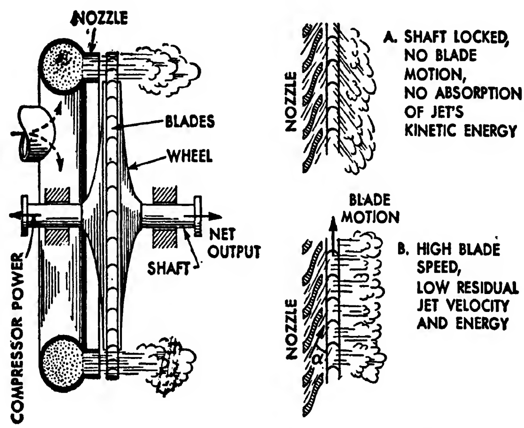


FIG. 10–1. Principle of the gas turbine.

to the shaft. The prospective jet speed is sometimes sufficiently high to warrant dividing the expansion into a series of *stages* with a set of nozzles and a row of blades in each stage, all blade wheels being mounted on the same shaft. By limiting the gas expansion per stage, the blade speed and revolutions per minute of the shaft are suitably decreased. Were the blades themselves so shaped as to be virtual nozzles, some expansion would also take place in the gas as it went through the blading. The latter would in consequence receive a "reaction thrust" distinct from impulse action. Many gas turbine designs have employed the reaction principle.

One to three stages in impulse types, and four to seven in reaction types, cover the multi-staging designs which have so far been published. The fluid mechanics of blade action in the steam turbine will be developed in Chapter 14, and as the fundamental principles apply to any case of compressible fluid flow, these details of blade action are not duplicated in this chapter.

A properly designed nozzle can produce almost an ideal (isentropic) adiabatic expansion of the gas. Any failure of the gas turbine to convert all the ideal available energy into work at the shaft is mainly attributable to the blading—its clearance, friction, leakage, and residual gas velocity.

The term "engine efficiency" is frequently applied both to engine and turbine prime movers to denote perfection of thermodynamic design, using the ideal available energy as a standard. So, if e_i is the ideal amount of energy made available by the possibility of expanding a fluid between specified initial and final states, and e is the energy actually developed by a gas turbine working between the same limits, then

$$\eta_e = e/e_i$$

where η_e is the so-called *engine efficiency*, sometimes called internal efficiency.

It is quite essential to the success of a gas turbine plant that η_e be as high as possible. The range of η_e is 80% to 90%, a remarkable modern achievement. Had this efficiency been better in early experimental gas turbine plants, some of the disappointments might possibly have turned out successfully.

Some conception of the relative positions of nozzles and blades may be gained from Figure 10–1. A single-stage impulse gas turbine is seen to be a shaft-mounted disk carrying on its rim a number of curved blades uniformly spaced around the periphery. The shaft is supported by two bearings, and terminates in flanges to which driven machinery may be coupled. Compressed gas is continuously supplied to a distributing ring, and thence to the nozzles, of which there may be several acting in parallel. The action of a nozzle in converting heat energy to kinetic energy of a jet was discussed in Chapter 4. Auxiliary figures 10–1A and 10–1B show that the axis of the nozzle actually makes an angle with the direction of blade travel. These figures may be thought of as a section of the curved rim, with its blades, as it would look if laid out in a horizontal plane.* With gas supplied to the nozzles at a pressure above that surrounding the wheel, an expansion will occur and the gas will be jetted against the wheel at angle α. If the blades were stationary (as is the premise of Figure 10–1A), the gas jet would shoot through the spaces between blades and emerge substantially undiminished in speed, although deflected in direction. Practically all the original kinetic energy would be retained by the jet. On the other hand, if the blades are moving their effect in deflecting the jet will be less, the jet energy will be partly absorbed as moving pressures against the blades.† By proper combination of blade speed, jet speed, α, and blade shape, the gas may be caused to transfer 80% to 90% of its kinetic energy to the wheel and to leave the wheel moving approximately parallel to the shaft.

This is the energy action of the gas turbine. A distinction must be made

* The auxiliary figures could be considered to be a *development* of the nozzles and blades.

† Remember that unless a force is allowed motion in the direction of the force, no work is involved. Chapter 1.

between a "gas turbine" and a "gas turbine plant." The gas turbine of Figure 10–1 assumes a steady and satisfactory supply of compressed gas to the nozzles. The gas turbine plant includes not only the turbine but also a means of producing a supply of gas at a suitable state.

Gas turbines present no new difficult problems to the manufacturer. For the same inlet temperature the gas turbine is probably easier of design than

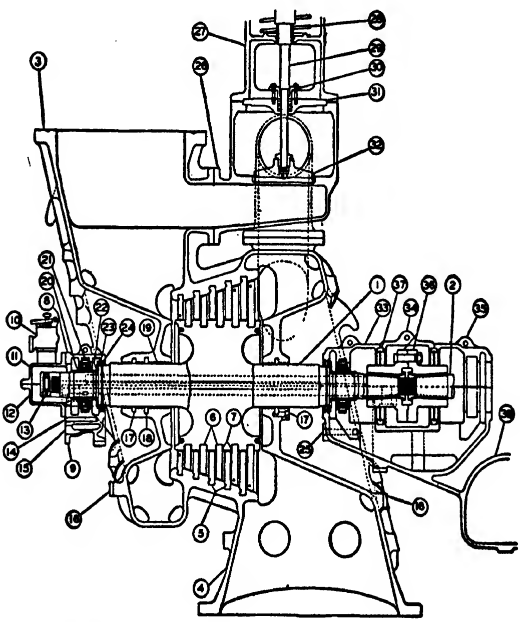


FIG. 10–2. Section through a five-stage reaction gas turbine. 1. Turbine spindle. 3. Inlet. 4. Exhaust. 5. Fixed blades (nozzles). 6. Moving blades. 10. Emergency overspeed govenor. 13. Emergency overspeed trip. 17. Sealing air. 19. Labyrinth seal. 21. Bearing. 26. Inlet to exhaust by-pass. 32. By-pass valve. Overspeed control. 36. Coupling to compressor. (Courtesy Allis Chalmers Mfg. Co.)

the steam turbine. Compared to the internal combustion engine, the turbine is compact and high speed. Gas turbine speeds of 10,000 rpm. are not uncommon. But because of the volume of the compressor and combustion chamber, and because much of the turbine output is absorbed in the plant itself, the advantage of compact size is not so likely to be realized, and often it is the internal combustion engine that is the more compact. However, in *simplicity*, the gas turbine excels.

A section through the gas turbine for a 40,000 cu. ft. per min. Houdry process unit is shown in Figure 10–2. Neglecting the numerous details, reference will be made to only the major parts.

A solid chrome-nickel-steel forging comprises the turbine spindle (1) on whose periphery are inserted five rows of reaction blading (6).

The turbine casing, split on the horizontal center line, is made of carbon-molybdenum steel. The gas inlet passage (3) and exhaust passage (4), both located in the same vertical plane, are cast integrally with their respective cylinder halves. Five rows of reaction cylinder blading (5) are inserted in grooves machined in the casing. A by-pass connection (26) joins the inlet and exhaust regions through a valve (32), operated by an emergency governor (10) which is actuated by a centrifugal stop bolt (13) when overspeed conditions exist.

Labyrinth glands (19) are provided to prevent leakage of gas to atmosphere where the turbine spindle ends emerge from the casing. To reduce gas

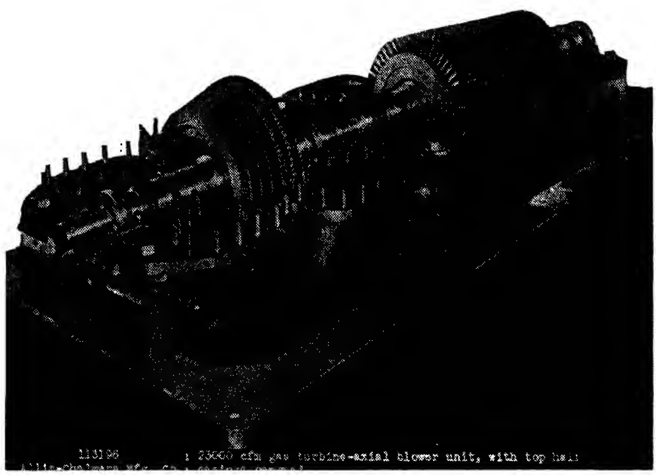


FIG. 10–3. Axial compressor unit for 23,000 cfm gas flow. Top half of casing removed. Turbine unit is in the foreground. (Courtesy Allis Chalmers Mfg. Co.)

leakage further, sealing air from the compressor discharge is injected at a suitable point (17) in the glands.

Conforming to certain requirements of the oil refinery units are the roller bearings (21) shown; however, these would be of the sleeve type in a machine designed for power purposes only. A solid coupling (36) joins the turbine spindle (1) with the rotor (2) of the axial compressor.

Example 1: A pound of gas with energy represented by an enthalpy of 273 B.t.u. per lb. is efficiently expanded through a group of gas nozzles to an enthalpy of 128 B.t.u. per lb. It is then directed into a row of blades which "wring" 86% of its energy from it. What are the gas velocities before and after contact with the blades? Also, if the blade speed is required to be one-half the initial gas speed in order to achieve this efficiency, and if the mean radius of the blade circle is 14 in., what are the necessary turbine rpm.?

The liberated heat ($273 - 128 = 145$ B.t.u.) is considered to be entirely converted into kinetic energy. Therefore the initial jet energy is 145×778, or 112,800 ft. lbs. In the kinetic energy form this is $\frac{1}{2}mv^2$, where $m = 1/32.2$ slugs, and $v =$ jet speed.

Solving for v,

$$\text{Initial } v = \sqrt{\frac{112{,}800}{\frac{1}{2} \times \frac{1}{32.2}}} = 2700 \text{ ft. per sec.}$$

The final gas speed, assuming no irreversible heating from friction or turbulence, is found by expressing the residual energy in the kinetic form. Residual energy = 112,800(1.00 − .86) B.t.u. per lb.

$$\text{Final } v = \sqrt{\frac{112{,}800 \times .14}{\frac{1}{2} \times \frac{1}{32.2}}} = 1000 \text{ ft. per sec.}$$

Using the assumed data, blade speed = $\frac{1}{2} \times 2700 = 1350$ ft. per sec.

$$\text{Blade speed} = 2\pi \times \text{revolutions per sec.} \times \text{radius of blade circle.}$$

So,

$$\text{Turbine speed} = \frac{1350}{2\pi \times \left(\frac{14}{12}\right)} = 184 \text{ revolutions per sec.}$$

This is 11,000 rpm. This extremely high speed may be accommodated in some cases, as in engine superchargers and jet aircraft engines, or where reduction gears are contemplated, but is impractical for numerous cases of direct drive. This example furnishes a clue to one of the more pressing reasons for adopting the complication of multistaging.

Example 2: An impulse type gas turbine receives 20 lbs. of gas per sec. at a temperature of 1100° F. It expands this gas to 550° F in three stages, each one of which is composed of a set of nozzles and a disk mounting a row of blades. Assuming ideal conditions, also that equal enthalpy quantities are released in each stage, and best blade speed is half the jet speed, find required wheel diameters for 5000 rpm., also the approximate power rating.

Change in enthalpy of a gas is $c_p \Delta T$. Assuming this is air, an average c_p is .245. Enthalpy change per stage = $\frac{1}{3}[.245(1100 - 550)] = 45$ B.t.u. per lb. of air. This is capable of producing a jet speed of

$$v = \sqrt{\frac{45 \times 778}{\frac{1}{2} \times \frac{1}{32.2}}} = 1500 \text{ ft. per sec.}$$

Blade speed = 750 ft. per sec.
Shaft speed = 5000/60 = 83.3 revolutions per sec.

Mean diameter of blade circle $= \dfrac{750}{\pi \times 83.3} = 2.86$ ft. (34.4 in.).

Enthalpy change per min. in the turbine = $20 \times 60 \times (45 \times 3) = 162{,}000$ B.t.u.
= 126,000,000 ft. lb. per min.

Power = 126,000,000/33,000 = 3800 hp. ideally.

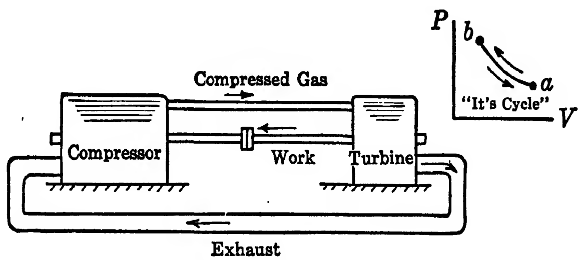
As gas turbines may have engine efficiencies of the magnitude of 80%, the actual output would approximate 3000 hp.

10–3. Gas Turbine Plant. In some instances, especially when a compact mobile unit is referred to, a plant is simply described as a "gas turbine." However, this book preserves a distinction. Whether a stationary or mobile plant is considered, the turbine is only one of the units that comprise it. The others are compressor, combustion chamber, controls, etc. In a few instances a supply of gas suitable for use in a gas turbine is created by an industrial process. Then, if in sufficient quantity, it can be utilized in a gas turbine. However, the greatest interest centers at present in the gas turbine plant which self-sufficiently will manufacture its own supply of gas by compressing air to a suitable pressure and burning some fuel in it.

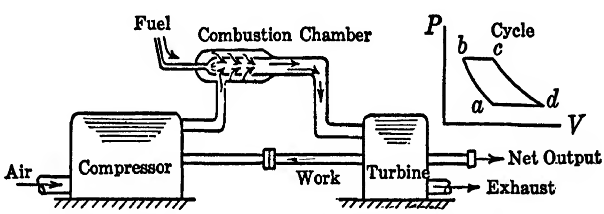
As was noted in a preceding section, the nozzles must receive the gas at a pressure above that of the surrounding atmosphere in order to achieve an expansion. In a gas turbine *plant*, then, there will be an air compressor. Ideally the turbine might power the compressor and receive from the compressor the gas to expand in the turbine nozzles. If the internal efficiencies of both machines were 100% (an impossibility, since among other things, it implies frictionless flow), all that is produced is a perpetual motion machine of zero net output. Upon assuming frictionless conditions, simpler designs of perpetual motion machines than this can be imagined. So, the simple turbine-compressor pairing pictured in Figure 10–4A is of no avail in producing power. It is equivalent to the mythical act of "raising oneself by one's own boot-straps." Since actual internal efficiencies are less than 100%, this "plant" would require an input of energy just to continue in motion. However, if between compressor and turbine some fuel is burned in the compressed air, it will *expand* to a larger volume before entering the turbine, and will be capable of performing more work than was required to compress the air. If this difference is enough to overcome the internal imperfections of both turbine and compressor, surplus work will become available as a *net plant output*. This net plant output (in B.t.u.), divided by the heating value of the fuel employed to produce it, is the overall thermal efficiency of the gas turbine plant. An elementary plant of this type is shown in Figure 10–4B. The working medium is changed in chemical composition in the combustion chamber, hence it is jettisoned after giving up energy in the turbine, and fresh air is drawn into the compressor. The cycle is therefore an "open" one. However, the cycle diagram is drawn as if the air passed around a closed cycle and received heat, only, in the combustion chamber. The line a–b represents compression of the air. The process b–c which takes place in the combustion chamber is an addition of heat at constant pressure. Expansion in the turbine is represented by c–d, while d–a is cooling the turbine exhaust to ambient atmospheric temperature. It is this part of the cycle that is open in gas turbine plants.

The constant pressure combustion cycle is used in most gas turbine plants. Designating it the *combustion gas turbine cycle* is a means of distinguishing

between it and the *explosion gas turbine cycle.* The latter type has been successfully built, but has the disadvantages of requiring a number of mechanically operated valves (like the I.C. engine), and of operating under fluctuating gas pressure. Explosion gas turbine plants (Holzwarth) have a combustion chamber into which air and fuel are admitted. The chamber is then closed, and the charge fired by a timed spark. Combustion takes place at constant volume. After combustion is completed the pressure in the chamber



A. Illustrating an unworkable arrangement even if the machines are theoretically perfect. Compare with B.



B. A workable gas turbine plant. Heat energy is added to increase the volume of gas entering the turbine.

FIG. 10–4. Basic combustion gas turbine plant.

is several atmospheres. The products of combustion are then released to the turbine nozzles. After the combustion chamber pressure is sufficiently dissipated, the cycle can be repeated. On account of its mechanical complexity, the explosion cycle is but little used.

Any liquid fuel which does not form corrosive products of combustion can be used in the open cycle plant. Since the products of combustion actually enter the turbine, ash content is not yet permissible, although experiments are under way on ultra fine coal pulverization in the hope that extremely small ash particles can be successfully passed through turbines. Refined fuel oil and kerosene are used except where an industry may have a gaseous fuel by-product, or where natural gas is available.

10–4. Cycle Efficiency. The air standard thermal efficiency of the basic combustion gas cycle *abcd*, Figure 10–4B, surprisingly enough turns out to be of the same form as the Otto cycle.

For the ideal basic gas turbine cycle,

$$\eta_t = 1 - \frac{1}{r^{\gamma-1}},$$

in which η_t = Air standard thermal efficiency.
r = Compression ratio V_a/V_b.
γ = Adiabatic polytropic exponent.

In flow machines such as the rotary compressor, the pressure ratio P_b/P_a is more useful than the volume ratio r. During adiabatic compression $P_b/P_a = r^\gamma$, hence

$$\eta_t = 1 - \frac{1}{\left(\frac{P_b}{P_a}\right)^{(\gamma-1)/\gamma}}.$$

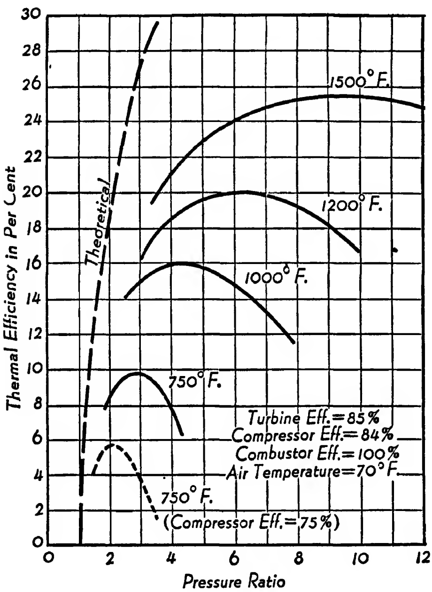


FIG. 10–5. Influence of variables on practical cycle efficiency. (Courtesy Westinghouse Electric & Mfg. Co.)

This equation shows that an increase in the pressure ratio will increase the ideal cycle efficiency. The actual cycle efficiency involves the internal efficiencies of turbine and compressor, as well as a variable air-fuel ratio, since for a given maximum cycle temperature, the higher the pressure ratio the greater the air-fuel ratio that must be employed. Consequently, for a limiting maximum temperature in the combustion chamber, the efficiency increases with pressure ratio up to a certain *optimum ratio*, then decreases as the pressure ratio is further increased. This is shown in Figure 10–5.

The thermal efficiency of the actual cycle may be approximated * as follows. Let W_c be the ideal work required in adiabatic compression of one pound of air and η_c be the internal efficiency of the compressor, based on isentropic compression. Then the actual input to the compressor is W_c/η_c. Likewise let W_t be the ideal work released in the turbine per pound of air (a simplifying assumption, because the turbine expands a mixture of air and products of combustion) and η_e be the turbine engine efficiency. The actual turbine output is $W_t\eta_e$. Call the heat added by the fuel per pound of air in

* A few sources of inefficiency such as pressure drops due to pipe friction are omitted in this analysis.

the combustion chamber, Q_f. The actual combustion gas turbine plant efficiency is

$$\eta = \frac{1}{J} \times \frac{\text{Net work done}}{\text{Heat added}} = \frac{W_t\eta_e - W_c/\eta_c}{778\ Q_f}$$

or

$$\eta = \frac{W_t\eta_e\eta_c - W_c}{778\eta_c Q_f}.$$

From page 138, W for adiabatic air compression in steady flow has the form:

$$W = \left(\frac{\gamma}{\gamma - 1}\right) wRT_1 \left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right].$$

If a suitable numerical value can be assigned to γ a solution may be made for W_t and W_c. Table 2–1 shows $\gamma = 1.4$ for air (also $c_p = .24$), but these values are for "moderate" temperatures. Moderate temperatures exist in the compressor, but γ for the turbine expansion will be less than 1.4 because (1) specific heats are not constant but change with temperature, (2) the working medium in the turbine is air diluted with products of combustion. It is not considered practical to investigate the field of variable specific heats in this book, but some allowance may be made by modifying values from Table 2–1. The constant γ will be used as 1.39 for the turbine; also c_p will be raised from .24 to .25 for average combustion chamber conditions.

Example 1: What is the efficiency of the basic gas turbine plant for the following data? Atmospheric conditions 14.7 psi. 60° F. η_c 80%, η_e 85%, pressure ratio 4, gases raised to 1000° F in the combustion chamber. Exhaust gas pressure 20 psi.

Writing an equation for work of the compressor per lb. air, in terms of the nomenclature of Figure 10-4B,

$$W_c = \frac{R\gamma T_a}{\gamma - 1}\left[\left(\frac{P_b}{P_a}\right)^{(\gamma-1)/\gamma} - 1\right] \text{ ft. lbs.}$$

in which R = Gas constant (Table 2–1).
T_a = Atmospheric temperature (absolute).
Other symbols as previously quoted.

The ideal turbine work per pound of air may be found from the same equation, with suitable rearrangement of subscripts, since the turbine expansion is equivalent to a reversed compression. It can also be analyzed that the turbine work is the flow work of entrance P_cV_c plus the nonflow work of expansion $\frac{P_cV_c - P_dV_d}{\gamma - 1}$* less the flow work of discharge P_dV_d

$$W_t = P_cV_c - P_dV_d + \frac{P_cV_c - P_dV_d}{\gamma - 1} = \frac{\gamma}{\gamma - 1}(P_cV_c - P_dV_d).$$

* See page 23.

Since $PV = RT$ (for 1 lb.)

$$W_t = \frac{\gamma R}{\gamma - 1}(T_c - T_d).$$

The ideal work done in compressing a pound of air, using $\gamma = 1.4$, is

$$W_c = \frac{53.4 \times 520° \times 1.4}{.4}[(4)^{.4/1.4} - 1] = 47{,}600 \text{ ft. lb.}$$

Assuming a datum of 60° F, the enthalpy of a pound of air leaving the compressor and (entering the combustion chamber) is

$$\frac{W_c}{\eta_c} = \frac{47{,}600}{778 \times .80} = 76.3 \text{ B.t.u.}$$

At 1000° F the enthalpy above 60° is $c_p(t_c - t_{\text{datum}}) = .25(1000 - 60) = 235$ B.t.u. Heat added in the combustion chamber $= 235 - 76.3 = 158.7$ B.t.u. per lb. To obtain T_d the general gas law $PV = \omega RT$ and the adiabatic equation $PV^\gamma = C$ are combined to yield

$$\frac{T_c}{T_d} = \left(\frac{P_c}{P_d}\right)^{(\gamma-1)/\gamma}$$

$$\frac{1000 + 460}{T_d} = \left(\frac{4 \times 14.7 \times 144}{20 \times 144}\right)^{(1.39-1)/1.39} = 2.94^{.28}.$$

$$T_d = 1080°.$$

$$T_c - T_d = 1460 - 1080 = 380° \text{ F.}$$

Substituting in the equation for W_t,

$$W_t = \frac{53.4 \times 1.39}{.39} \times 380 = 72{,}400 \text{ ft. lbs.}$$

Hence the overall plant efficiency is

$$\eta = \frac{72{,}400 \times .85 \times .80 - 47{,}600}{.80 \times 158.7 \times 778} = .017$$

$$= 1.7\%.$$

If, on account of combustion inefficiency, the 158.7 B.t.u. used above represented less than the lower heating value of the fuel involved, the efficiency would be still smaller.

In spite of the comparatively high engine and compression efficiency chosen, the overall efficiency is quite low. But it is to be noted that the two terms of the numerator are of approximately the same magnitude, therefore small changes of η_e or η_c will cause a large change in η. This solution also emphasizes the necessity of building the turbine for complete expansion to atmospheric pressure, for had 14.7 psi. been used in the solution in place of 20 psi. exhaust pressure, the exhaust temperature would have been reduced to 990° R and the efficiency raised to 13%.

Example 2: Find the *air rate*, using the data of Example 1. The air rate is defined as the pounds of air which must be compressed to provide a horsepower hour of net output.

From Example 1, the net output per pound of air is

$$72{,}400 \times .85 - \frac{47{,}600}{.80} = 2000 \text{ ft. lbs. per lb. of air.}$$

A hp. hr. is $33{,}000 \times 60 = 1{,}980{,}000$ ft. lbs.

$$\text{Air rate} = \frac{1{,}980{,}000}{2000} = 990 \text{ lbs. per hp. hr.}$$

By building the turbine for 14.7 exhaust, the air rate is reduced to 118 lbs. per hp. hr. Not only does the complete expansion provide a major increase in economy, but it also reduces the necessary bulk of equipment for a specified power rating.

10–5. Thermodynamic Refinement. The data obtained in the foregoing example might well dash the spirits of those who held hopes of the combustion gas turbine actively competing with the steam and internal combustion prime movers. Although nearly perfect operation is assumed for the machines, the efficiency is so low that our gas turbine is worsted by everything except the non-condensing steam plant. However there are possibilities of improving the performance through metallurgical progress permitting higher maximum temperatures (1500° F is being contemplated), and by thermodynamic schemes that reduce compressor demands and salvage part of the exhaust heat. Pushing up the maximum temperature limits will be a slow painstaking process, with progress dependent on developments in other fields of technology. But the thermodynamic modifications, called *intercooling*, *regeneration*, and *reheating*, may be applied at present. At the cost of complicating the flow pattern of a plant, whose claimed virtue has always been simplicity, efficiencies may be raised until this prime mover becomes a practical competitor with the others, for overall efficiencies of 20% to 30% seem to be possible, using today's maximum practical temperatures.

The intercooling principle consists of compounding the compression, and intercooling between sections. This lowers the air temperature at constant pressure, reducing the volume and the compression work required in the high-pressure compressor stages. It requires a surface type intercooler with a coolant, air or water. Regeneration is effected by transferring heat energy from the turbine exhaust gases to the compressed air after it leaves the compressor, but before it enters the combustion chamber. This again requires a surface heat exchanger, one capable of standing up under high temperatures. The gain here is due to the smaller quantity of fuel required to be injected to reach the same maximum gas temperature. Reheating is the process of burning the fuel in steps, the amount of fuel being proportioned to reach the

maximum allowable temperature in each stage of combustion. For this the turbine must be compounded, with a reheating combustion chamber inserted in the flow between the high- and low-pressure sections. These principles are illustrated diagrammatically in Figure 10–6. A marine gas turbine plant

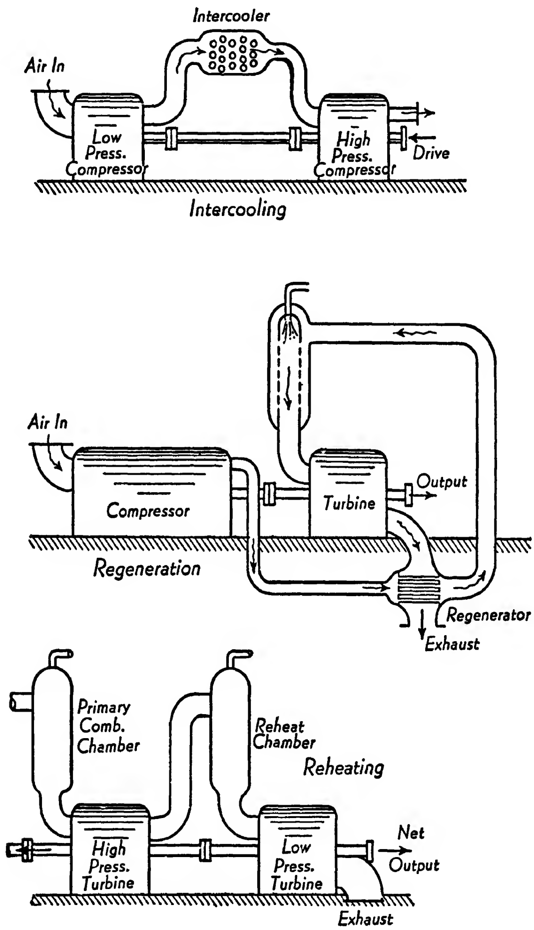


FIG. 10–6. Refinements of the combustion gas turbine plant.

recently built, and incorporating intercooling, regeneration, and reheat, is shown schematically in Figure 10–7, with attendant temperatures noted. This plant is characterized by the use of positive displacement rotary compressors rather than the axial flow type usually encountered. With a pressure ratio of 6.5 and a maximum throttle temperature of 1230° F, it is said to achieve an over all efficiency of 29% when producing the rated net output of 2500 hp. The high-pressure section of the turbine drives the low-pressure section of the compressor, these being proportioned so that there is no surplus power. After intercooling, the air is further compressed in the high-pressure

compressor, then heated by being passed through the regenerator. Fuel is next injected into it and burned. Plenty of oxygen remains in the products due to the great quantity of excess air used to hold down the temperature rise. After expansion through the high-pressure turbine more fuel is burned with some of the remaining oxygen, again raising the temperature to the limiting value. Next the products expand through the low-pressure turbine, after which they pass into the regenerator and thence to the exhaust stack. As this was designed for marine service, the net output of the low-pressure turbine is delivered to the propeller shaft at the output coupling of a speed-reducing gear box.

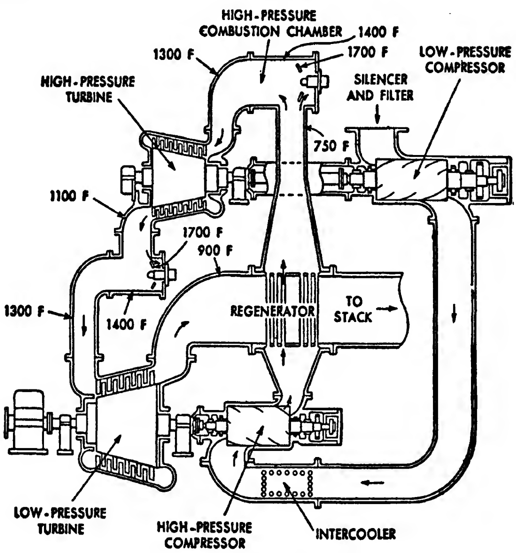


FIG. 10-7. An intercooled, regenerative-reheating gas turbine plant in schematic arrangement. (Courtesy Elliott Co.)

It is to be expected that all these improvements are not always justifiable economically. The cheaper the fuel the fewer the allowable efficiency-building modifications. Regeneration, especially, is expensive because of the rigorous service conditions of the heat transfer surface and the large area needed to obtain "effective" regeneration.* A 10,000 kw. stationary gas turbine plant using low-cost natural gas was built to realize 21% thermal efficiency using intercooling and reheating, but omitting regeneration. With a regenerator the efficiency might have been raised to 30%, but comparative costs ruled out the improvement.

* The *effectiveness* of gas heat transfer surfaces is sometimes described by the ratio of temperature drop of the heat releasing gas to the maximum difference in temperature between the two gases. To obtain effectiveness of 50% or more, regeneration surface of 4 to 10 sq. ft. per kw. capacity is needed. This is quite bulky compared to the 1 sq. ft. in steam condensers.

10–6. Gas Turbine Plant Compressors. Although the compression pressures required in this type of power plant are not considered to be "high," such large volumes are involved that only steady flow compressors are economic. Reciprocating compressors were used on some of the earlier experimental models, but currently only rotary compressors are considered. These may be of the following types:

1. *Centrifugal.* Characteristics are very high rotative speed, single-stage, moderate pressure ratios. This type is found principally in the turbo-supercharger and jet engine fields, where other equipment (generators) is not driven.

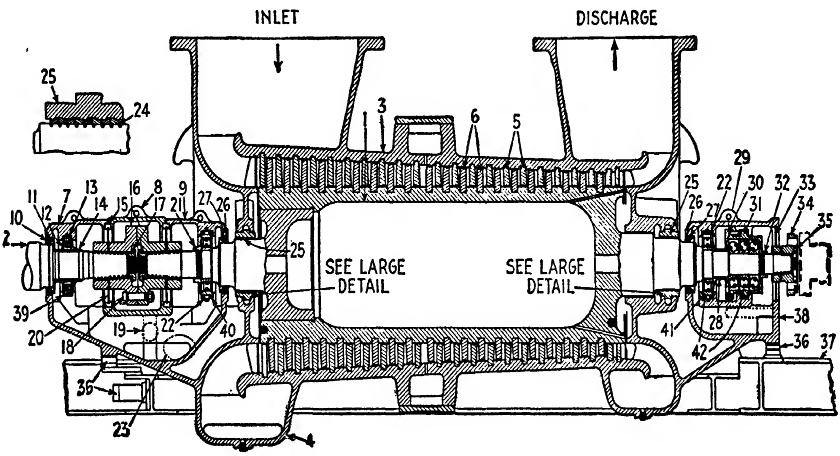


FIG. 10–8. Section through an axial flow compressor. (Courtesy Allis Chalmers Mfg. Co.)

2. *Rotary Positive Displacement.* These are generally called rotary blowers or rotary compressors. These have the advantage of positive displacement and do not require high rotative speed for building up high-pressure ratios. They are highly exacting in the requirements for machine precision during manufacture. Although generally employed as a gear-driven supercharger for relatively slow-speed Diesel engines, at least one gas turbine plant builder has used this type of compressor.

3. *Axial Flow Compressor.* Not much attention had been devoted to this type of compressor before its adaptation to the gas turbine plant. However, a decade of development in this field has brought it to a highly perfected state. It is by nature a gas turbine running reversed. Its principle of action was mentioned in Chapter 6. By the use of a large number of stages the pressure can be built up, efficiently, to as much as 60 psi. gage pressure. Figure 10–8 shows, in a long sectional view, the typical construction of one of these compressors. Each stage consists of a row of stationary and a row of moving blades, the latter being mounted on a hollow drum. The blades are tightly secured in grooves machined on the outside of the drum. The drum termi-

nates endwise in shafts which support it in two bearings. As the air decreases in volume during compression, the blade heights become progressively lower from inlet to outlet. The axial thrust generated by the air flow is opposed and neutralized by the turbine axial thrust when the two shafts are coupled.

10–7. Combustion Chambers. Gas turbine plant fuels are gaseous or liquid. In either case the fuel can be injected into the combustion chamber only after attaining a pressure exceeding that of the compressor discharge. Liquid fuels are readily injected with a pressure pump, connected to a continuous-flow injection nozzle, but some considerable amount of work might

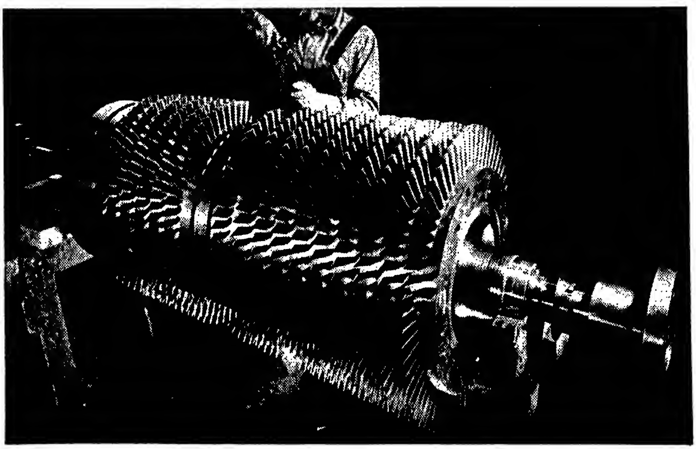


FIG. 10–9. Roter of an axial flow compressor being bladed. (Courtesy Allis Chalmers Mfg. Co.)

be consumed in compressing a lean gaseous fuel such as blast furnace gas. However, natural gas is frequently available at pressures high enough to cause its flow into the combustion chamber without further compression.

To maintain the products of combustion at temperatures safe for turbine operation a great deal of excess air must be furnished. The cooling effect of all this air mitigates against best combustion conditions existing at the burner, so often only a part of it is sent to the burner. The remainder is mixed with the products of combustion in order to cool them sufficiently before entering the turbine. This air may also be employed to jacket and protect the burner or combustion chamber walls in the region of maximum temperature.

10–8. Applications of the Combustion Gas Turbine. Advocates of gas turbines hope that, ultimately, this form of prime mover will come into active competition with the Diesel engine and the steam turbine for regular service in stationary power plants. Much remains to be learned about the reliability and stamina of these plants; but it seems that they may be built with thermal efficiencies which are comparable to the older types. Simple combustion gas

turbine plants have already been built for stand-by service. Some highly efficient units may be expected to appear in regular service in the future.

An experimental locomotive unit was successful but not particularly impressive in performance. However, subsequent progress may raise performance to competitive levels. A gas turbine locomotive is smooth in action and easier on the roadbed than the reciprocating steam engine locomotive. It needs no large quantity of water, as does the latter, and is said to be considerably lighter per horsepower. The steam turbine locomotive is now being tried out in the United States. Undoubtedly an experimental gas turbine locomotive will sooner or later also be placed in service.

FIG. 10–10. Foreign gas turbine plant. 4000 kw. generator in foreground. (Courtesy Allis-Chalmers Mfg. Co.)

Most of the gas turbines manufactured to date in this country were produced for use with the Houdry process. In this patented gasoline cracking process compressed air must be provided for burning deposits from the catalyst. This act generates a considerable rise of temperature, and takes place with only a small drop of pressure. As a by-product, then, there is available a quantity of compressed gas. This gas is put through a gas turbine whose output supplies the power necessary to compress air for the process. Here no net power is made, but since otherwise some 2500 hp. would be required to drive the Houdry process compressors, the turbine-compressor combination is highly practical.

The gas-turbine-driven internal combustion engine supercharger has been mentioned in Chapters 8 and 9. The performance of the high-capacity aeronautical engine when equipped with a turbo-supercharger excels at extreme altitudes. It is likely that this will remain principally in the military field, as the needs of commercial aviation are well met by the gear-driven super-

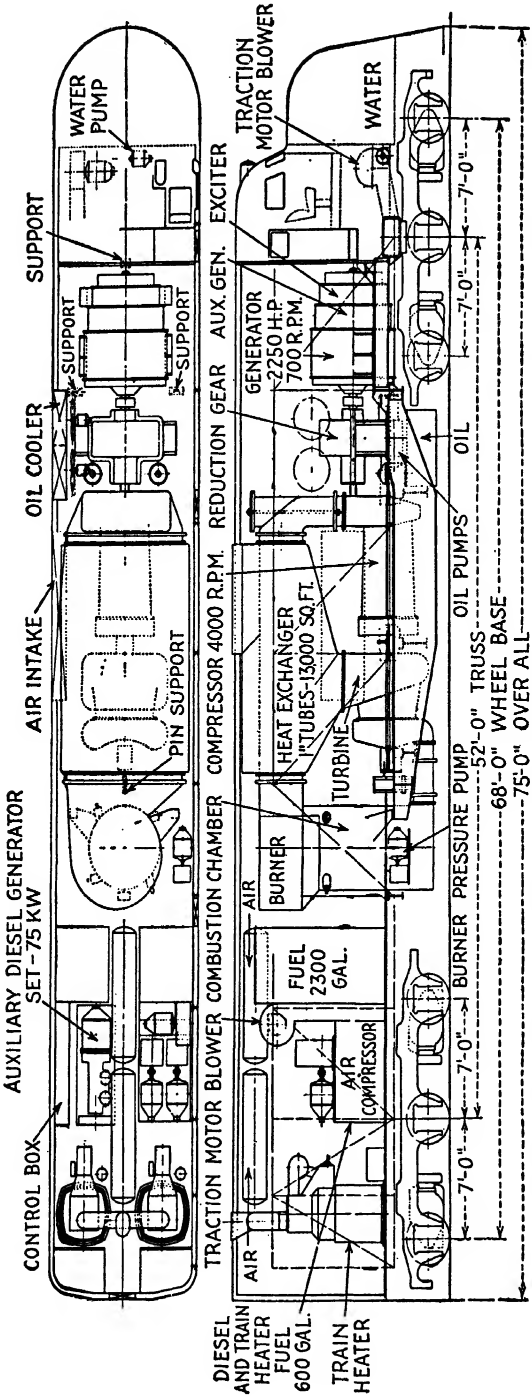


FIG. 10–11. Arrangement diagram of 2250 hp. combustion turbine locomotive with electric drive. (Courtesy Allis Chalmers Mfg. Co.)

charger, except possibly for airliners designed for the stratosphere. On the other hand, many commercial Diesels, especially where compact design is sought through the use of high cylinder mean effective pressures, are advantageously supercharged with gas-turbine-driven centrifugal superchargers.

Figure 10–13 shows an aviation type turbo-supercharger. Exhaust gases enter a circular casing and are discharged downward through nozzles which form the gas outlet from the casing. The gas liberated through the nozzles

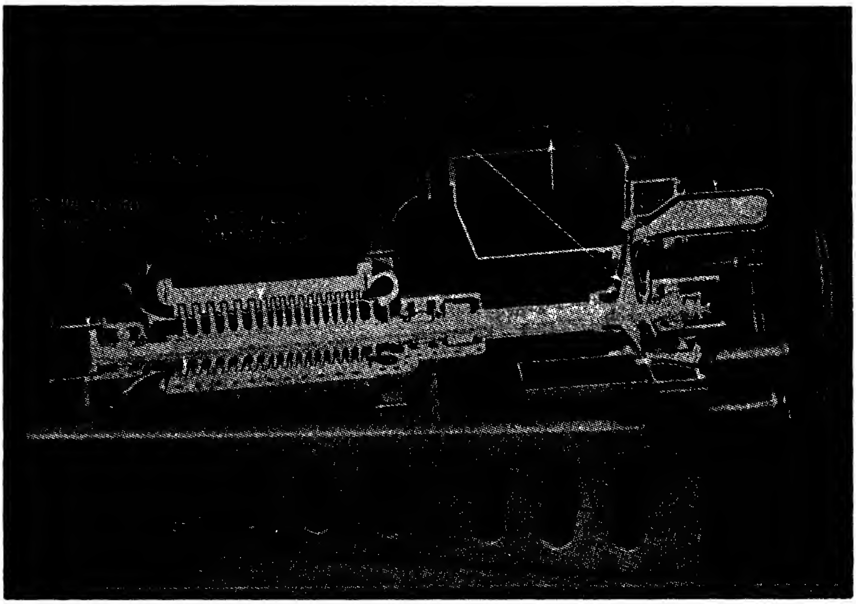


FIG. 10–12. Wooden model of gas-turbine power plant (for marine application) without removable portion—to show apparatus in section.

strikes the blades of the turbine, spinning the wheel several thousand revolutions per minute. Emerging from the blading, the gas is free in the atmosphere. All the turbine output is absorbed by the impeller of the centrifugal compressor, and if more gas power is available than is needed, some is valved through the waste gate, for otherwise the wheel might overspeed.

The gas turbine offers efficiencies about equivalent to the usual marine power plant. If, as it seems possible, this type of plant can be made to occupy less space (it already is lighter in weight) than existing prime movers, an important field of application will be opened to it.

About the most recent application of gas turbines is to be found in the field of jet propulsion of airplanes. There the gas turbine is used to compress sufficient air for combustion to result in the formation of a large, high-speed, gas jet whose reaction provides the airplane propulsion instead of the usual airscrew. This equipment is mentioned in some detail in Chapter 11.

Summarizing the comparative features of the gas turbine as a prime mover:

1. Mechanically it is simple compared to steam and I.C. plants, but in the endeavor to reach competitive efficiencies some of this advantage is lost.
2. An electric motor or I.C. engine is required to start the gas turbine plant. As the starter must bring the compressor well up toward operating speed, starting is not as simple as S.I. engines, but many compare favorably with C.I. engines.
3. Like steam turbines, the gas turbine is not readily reversible. Steam engines and two-cycle I.C. engines are best in this respect.
4. Turbine plants have less vibration than engine plants of similar size.
5. The gas turbine uses high temperatures. Even though the pressures are moderate, service life will be shortened by high temperatures.
6. With certain types of compressors, efficiency of the gas turbine plant is not as well maintained at part load as with steam or the I.C. engine.

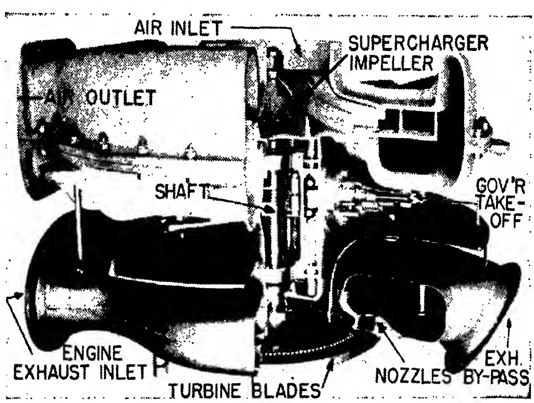


FIG. 10–13. Gas-turbine-driven internal combustion engine supercharger (aviation engine type). (Courtesy General Electric Co.)

PROBLEMS

1. Prove that the "engine efficiency" of a gas turbine equals the ratio of the actual to the ideal temperature drop.

2. A single-stage gas turbine will receive gas at 40 psi. gage, 1200° F, and develop 135 hp. per lb. of gas flowing per sec. $c_p = .25$, $\gamma = 1.39$, engine efficiency = 85%. Find (a) the actual exhaust gas temperature, (b) the ideal exhaust gas temperature, (c) the exhaust pressure, using $T_2/T_1 = (P_2/P_1)^{(\gamma-1)/\gamma}$.

3. The exhaust pressure of a single-stage gas turbine is 15 psi. abs. Combustion chamber conditions: 1100° F, 55 psi. gage. The *air rate* is 22.7 lbs. per gross hp. hr. Estimate the engine efficiency.

4. What speed would gas have after expanding from 50 psi. gage, 1230° F to atmospheric pressure, then giving up 88% of its kinetic energy in the turbine? If 80%

of the kinetic energy is converted into work and 8% is reabsorbed by the gas as heat, what is the final temperature? $\gamma = 1.39$, $c_p = .25$.

5. It is desirable to operate the blading of a single-stage turbine at half the speed the gas (air) has as it leaves the nozzles. Show that if the turbine rpm. is 2500 and the blade circle 14 in. in diameter, ideally a pressure ratio of only 1.02 is possible. Take initial temperature 1100° F. See Section 4–8 for nozzle equation.

6. Following the method of the sample examples of this chapter, endeavor to check the peak of the 1200° curve of Figure 10–5.

7. Following the method of the sample examples of this chapter, endeavor to check the peak of the 1500° curve of Figure 10–5.

8. Draw a sectional diagram of the turbine of Example 2, page 265, using information from Figure 10–2.

9. Assume that the turbine of Figure 10–2 was designed to have the gas yield up 15 B.t.u. per lb. as it passed each blade, fixed and moving. Considering the "engine efficiency" as 100%, what horsepower would be produced by a flow of 5 lbs. per sec.?

10. A closed ideal combustion gas turbine cycle is to be plotted from the following data, and for one pound of air. (See Figure 10–4B). $t_a = 60°$ F, $p_a = 14.7$ psi., $t_c = 1350°$ F, pressure ratio 4, $\gamma = 1.4$ throughout. Compute η_t, the theoretical efficiency of this cycle. Scales: 1 in. = 10 psi., 1 in. = 5 cu. ft.

11. Plot a cycle similar to that shown in Figure 10–4B for one pound of air, and calculate its theoretical efficiency. $t_a = 60°$ F, $p_a = 14.7$ psi., $t_c = 1200°$ F, pressure ratio 6. $\gamma = 1.4$ throughout. Scales: 1 in. = 10 psi., 1 in. = 4 cu. ft.

12. From the brief description contained in this chapter, construct a diagram of the equipment needed for an explosion gas turbine plant. Diagram its cycle on the P–V plane.

13. Compute the air standard efficiency of the closed combustion gas turbine cycle having pressure ratio of 6, using $\gamma = 1.4$.

14. What is the η_t efficiency of the ideal cycle, Figure 10–4B, for pressure ratio of 4? $\gamma = 1.4$.

15. Using the data of Problem 11, and $\eta_e = 79\%$, at what value of η_c will the combustion gas turbine plant just break even, i.e., have zero η efficiency?

16. Using the data of Problem 10, what product $\eta_e\eta_c$ will result in zero efficiency of the combustion gas turbine plant? Sketch a graph of η_e vs. η_c for a range up to 90% in either case. (Hint: Use equations, page 269, setting $\eta = 0$.)

17. Diagram the gas turbine plant of Figure 10–11 in a manner similar to that employed in Figure 10–4.

18. Diagram a gas turbine plant having intercooling and reheating.

19. Diagram a gas turbine plant having intercooling and regeneration.

20. Reconstruct the plant diagrammed in Figure 10–7, using equipment symbols like those of Figure 10–6.

21. Calculate the efficiency that the plant of Example 1, page 269, would have if regeneration of 50% "effectiveness" were included. c_p for air = .24, for products of combustion = .25.

22. A basic combustion gas turbine plant has a pressure ratio of 4. Atmospheric pressure 14.7 psi., temperature 60° F, turbine exhaust pressure 14.7 psi. Maximum temperature 1000° F, η_c 80%, η_e 85%. Combustion efficiency 100%. Calculate plant thermal efficiency and air rate.

23. A gas turbine plant with equipment as shown in Figure 10–4B operates under the following conditions: Maximum temperature 1200° F, pressure ratio *optimum.*

η_c 82%, η_e 84%, atmospheric condition 14.7 psi., 0° F. Turbine exhaust 16 psi. abs. Combustion efficiency 90%. Find the plant thermal efficiency and the air rate.

24. What would the efficiency rise to in the plant of Problem 23 if a regenerator of 50% "effectiveness" were added?

25. By what amount would the compressor work have been decreased if in Example 1, page 269, the air had been intercooled to 100° after compression to 35 psi. abs.? How much would this have improved the overall plant efficiency? Specify the required "effectiveness" of the intercooling surface.

CHAPTER 11

Jet Propulsion

11–1. Reaction Principle. Jet propulsion, as commonly practiced, is reaction propulsion. In the majority of cases where both action and reaction exist, it is the action that is sought for use. Then the reaction claims attention chiefly on account of the need for providing anchorage or support to the frame or foundation receiving it. But jet propulsion wastes the action and *uses the reaction.*

Principles governing jet propulsion belong in the field of dynamics. Classical dynamics rests upon the three Newtonian Laws, the third of which is the law of action-reaction. It states that for every action there is an equal and oppositely directed reaction. When a bridge rests on a pier and presses down against it with a force derived from the weight and load of the bridge, the pier simultaneously exerts an upward thrust of like magnitude against the bridge. This upward force is the "reaction." Jets which are accelerated out of a stationary nozzle produce a reaction on the nozzle that would drive it backward from the jet were it free to move. Where arrangements are made to use this jet reaction it can become a propellant for vehicles, bombs, etc.

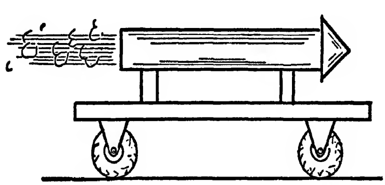


FIG. 11–1. Simple reaction propulsion.

Consider the case of a jet discharged from a body attached to a car, as shown by Figure 11–1. For those who need something concrete to visualize, it may be imagined that the source of this jet is a large skyrocket. If the car is light weight and equipped with anti-friction bearings, one can readily imagine that it would be accelerated forward speedily upon ignition of the rocket. Now many may possess a feeling that this, in some way, is because the jet from the rocket pushes against the air—something in the manner of gas in an engine cylinder pushing against the piston. They will doubtless be surprised to learn that the rocket would use its fuel even more efficiently if operated in the absence of an atmosphere—as in the outer regions of space. In any necessary adjustment of previous concepts, let it be remembered that in this case it is the reaction that becomes the useful force, and its existence is not contingent upon a surrounding atmosphere.

When the rocket contents burn, the products are expelled at high velocity and a forward thrust is imparted to the rocket case. If a constant rate of discharge of m slugs per second is assumed, and v is the velocity of discharge, relative to the rocket, which takes place through the nozzle mouth at section aa, Figure 11–2, then a reaction force is set up which is caused by the momentum change suffered by m. This force is mv. Due to the rocket's symmetrical shape, the air pressure acting externally on it is balanced save for the area at aa and an equal parallel one at the closed end of the rocket body. Let P_m be

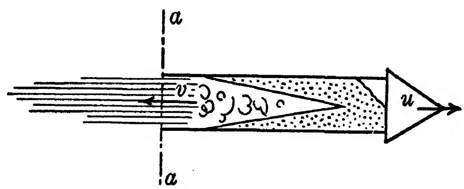


FIG. 11–2. Thrusting rocket.

the gas pressure at aa, A the area there, and P_1 atmospheric pressure. The net static pressure is $A(P_m - P_1)$ acting with mv.

$$F = mv + A(P_m - P_1) \text{ gross rocket thrust.}$$

The net thrust would require air drag to be deducted from this force, possibly also a gravitational component.

If the nozzle expands the jet to atmospheric pressure $P_m = P_1$, then $F = mv$. This is often cited as the thrust equation of true rockets. The same can be obtained from an energy analysis, for the original kinetic energy of mass mt, t being time, and the realized heating value Q that creates the velocity v in that mass are, together, equivalent to the thrust work Fut plus the residual energy existing at the final absolute speed of $v - u$. This, written symbolically thus,

$$\tfrac{1}{2}mtu^2 + \tfrac{1}{2}mtv^2 = Fut + \tfrac{1}{2}mt(v - u)^2$$

becomes, as before,

$$F = mv.$$

The efficiency of propulsion for a fixed jet velocity, v, and variable rocket speed, u, is given by the equation:

$$\eta = \frac{2uv}{v^2 + u^2}.$$

Hence it appears that, theoretically at least, the thrust remains constant, but efficiency of use of the energy developed by combustion increases with u until it becomes 100% when $u = v$, thereafter falling off as u is increased above v.

In a jet engine the fuel is carried in the vehicle which the engine propels, but oxygen is secured by taking in air from the surrounding atmosphere. As very high air-fuel ratios are employed in combustion jet engines, a good approximation, with simplifying features, is to neglect the mass of the fuel and consider that heat only is added to the air passing through the engine.

As the jet engine is diagrammed, much simplified, in Figure 11–3, it is illustrative of jet propulsion in any fluid which may somehow or other be accelerated as it passes through the engine. Although, in reality, the engine moves forward through a stationary fluid with a velocity of u, it is sometimes convenient to "invert" the relative velocities and consider the engine at rest in a fluid stream moving at velocity u past it. The region enclosed by the dotted boundary contains whatever means may be employed to accelerate the fluid flow from u to v. For example, this would consist of the compressor,

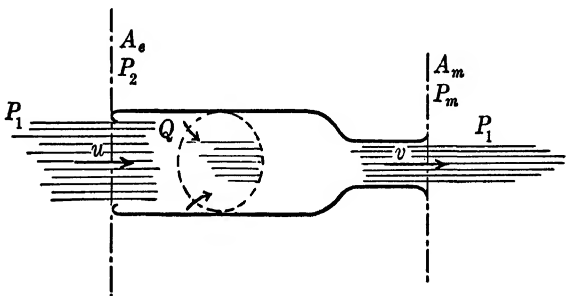


FIG. 11–3. Jet engine inversion.

burner, and exhaust nozzle of a turbo-jet airplane engine. Or it would be the fuel injection and combustion apparatus of the Athodyd, which is a simple but highly intriguing jet engine using only air ram for compression.

Jet propulsion is secured as follows from the compression jet engine. Let m be the mass per second of the air involved. From Figure 11–3 one visualizes the following action. Air rammed into the entrance scoop comes to rest producing a ram pressure, at the entrance, of P_2. It is then picked up by a compressor and given some appropriate compression. After fuel heat Q is added to it there is a velocity-producing expansion causing a jet to be expelled from the mouth of the nozzle at velocity v relative to the engine. Let P_m be the gas pressure at exit and A_m the area of the nozzle there. Also A_e = area of entrance, and P_1 = atmospheric pressure. The thrust $F = mv - mu + (P_mA_m - P_2A_e) + P_1(A_e - A_m)$.

When the engine is in motion through still air the energy it receives in t time is Fut. The residual, or unused, energy is $\frac{1}{2}mt(v - u)^2$. Efficiency being useful energy divided by useful energy plus losses,

$$\eta = \frac{Fut}{Fut + \frac{1}{2}mt(v - u)^2}.$$

Monographs by different authors are found to interpret this equation differently, depending on their assumptions regarding the PA terms of the expression (above) for thrust F. Assuming that $P_m = P_1 = P_2$ (ideally possible in a turbo-jet engine), then

$$\eta = \frac{2(uv - u^2)}{v^2 - u^2}.$$

In the case of Athodyd the engine reaction is principally an expansion of gases at constant (ram) pressure, making $v = u$ and causing A_m to exceed A_e.

No mechanical means of compression is furnished. If Athodyd has a speed of u, a dynamic pressure increment ΔP can be built up in the Athodyd body by air ram. Fuel is burned so that a velocity v equal to u will exist at the mouth. Since $v = u$ no momentum change is involved and thrust is derived from static pressure. The equation given above for F becomes $F = \Delta P(A_m - A_e)$ where $P_2 = P_1 + \Delta P = P_m$ and $v = u$. As residual velocity $(v - u)$ is zero, the efficiency of Athodyd, operating under the assumed conditions, is 100%.

The thrust of the compression jet engine is mv at standstill, and decreases by the term mu as velocity is increased. Athodyd has zero thrust until some dynamic pressure $\rho u^2/2$ exists.

It has been proposed that marine vessels, in particular submarines, be propelled by the reaction obtained by imparting a high velocity to a quantity of water taken in through a lengthwise duct, then discharged rearward.

Some cases illustrating reaction propulsion are:

1. The skyrocket.
2. War rockets (Bazooka, etc.).
3. Liquid fuel rockets (V-2).
4. Jet engines for airplanes.
5. Resonance jet (Buzz bomb).

11-2. Rockets. Rockets are reaction-propelled bodies which carry all the material from which the propulsion jet is formed along with them on their flight. In contrast, jet engines carry only fuel and rely on the surrounding atmosphere for oxygen to support combustion. Jet engines must remain in the atmosphere. Rockets travel best in the absence of an atmosphere; they are the only propulsion means yet developed which may ultimately solve the problem of the "space ship."

Rockets are to be classified according to the fuel type, and by their usage.

A. Type of fuel.
   1. Solid. Gunpowder, etc.
   2. Liquid. Alcohol, gasoline.

B. Usage.

1. Display and signalling. Skyrockets.
2. Life saving. Breeches buoy line carrier.
3. War rockets.
4. Thrust augmenter (rocket assist).
5. Rocket aircraft.
6. Meteorological exploration.
7. Possible future "space" rockets.

There have been several ill-considered attempts to employ rockets for purposes foreign to the nature of the rocket. Thus rocket-propelled automobiles are a dismal failure because the operating conditions will never permit the attainment of an efficiency approaching that of the internal combustion engine and other prime movers. Rockets are high-speed affairs. For them, fifty, a hundred, or two hundred miles an hour is a snail-pace. Until speeds higher than our fastest airplanes are exceeded, rockets remain comparatively inefficient.

Rockets are good for the creation of high velocities without recoil on the launching apparatus. This property gives them peculiar military significance. In rocket form heavy projectiles may readily be fired from small boats, aircraft, and by infantrymen, whereas the same projectile shot, as from artillery, would be accompanied by such a recoil that neither man, boat, nor aircraft could withstand it. So in World War II small landing craft used in amphibious attacks were packed with fire power equivalent, it was said, to the guns of a cruiser by mounting on them multiple war rocket launchers. Two-man teams delivered the wallop of a heavy piece of anti-tank ordnance with the "bazooka," a short-range high-velocity rocket of $3\frac{1}{2}$-lb. weight with high explosive war-head. Comparatively small warplanes carried several rockets mounted on launching rails usually attached below the wing. This gave them an offensive power nearly equivalent to the mounting of cannon in airplanes —a wartime project that was never completely successful. But portability and simplicity of launching the missile are the war rocket's chief advantages. In long-range aiming accuracy, and efficiency of use of the propellent, artillery is greatly superior.

As was previously noted, rockets are good for the creation of high velocities. The subsonic velocities of modern aircraft are too low for efficient rocket propulsion, although rocket planes had some wartime development. The limited duration of flight became a handicap, but the short bursts of exceptionally high speed had a tactical use in intercepter aircraft. While the rocket has been principally the child of warfare, scientists, explorers, astronomers, and other venturesome souls look with interest on what may next come from it, for the rocket is good principally as a vehicle to navigate empty

space. Here it will find the biggest field of all—in some history-making future hour.

No rocket is simpler in construction than the skyrocket (Figure 11-4). Powder is closely packed in a cardboard tube, leaving a clear cone-like space in the center to serve as a combustion chamber. A common gunpowder formula is charcoal (C) 15%, sulfur (S) 10%, and saltpeter (KNO_3) 75%. It is desirable to slow down the rate of combustion a little in rockets by using less of the oxidizing substance in the mixture. A typical rocket powder is KNO_3 56%, C 32%, S 12%. A clay nozzle is formed inside the rocket tube in order to form and guide the jet. When the powder is ignited by means of a fuse, combustion gases form quickly at a pressure in the central chamber and are violently expelled, carrying along bits of burning carbon. As the powder burns the combustion chamber volume increases and the rate of combustion also. The powder is quickly consumed, but the acceleration produced is quite high. A rocket weighing initially 2 lbs., charged with 0.7 lb. of powder, would produce an average thrust of 11 lbs. if it burned uniformly and had a jet velocity of 1500 ft. per sec. This can give the 2-lb. rocket an acceleration of $5.5g$. The velocity theoretically attained, neglecting air drag and gravitation, is obtained from the equation for momentum, and is

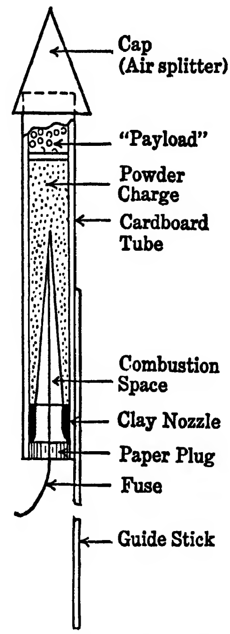


FIG. 11-4. Skyrocket.

$$u_2 = v \log_e \frac{M_1}{M_2},$$

in which v = Jet velocity relative to the rocket nozzle.
u_2 = Maximum velocity of the rocket.
M_1 = Initial rocket mass.
M_2 = Final rocket mass.
$M_1 - M_2$ = Weight of powder.
M_1/M_2 = "Mass ratio."

With the rocket quantities as mentioned above, the mass ratio is 2/1.3, with a natural logarithm of 0.432. So the ideal maximum velocity is 43.2% of jet velocity, or 650 ft. per sec. Were the rocket aimed for vertical ascent, part of the thrust would overcome gravitation, part would overcome air drag, and the remainder would accelerate, with a final speed probably not over 400 ft. per sec. Still, this is 270 mi. per hr.

Example: It is estimated that cordite used as a rocket propellant is capable of producing 2500 ft. per sec. exhaust velocity. Then what powder charge is necessary to project a rocket having a 2-lb. case and carrying 2 lbs. of explosive to a velocity of 1700 ft. per sec.?. Consider horizontal launching and neglect air resistance.

After the powder is spent the case and explosive, weighing 4 lbs., become the mass M_2. If u/v is to be $1700/2500 = 0.68$ and $\log_e M_1/M_2 = u/v$, then $M_1/M_2 = 1.974$ and M_1 is 7.8 lbs. The powder charge is $7.8-4.0$, or 3.8 lbs. If the powder burns uniformly over a period of two seconds (this depends on the detail design), then $m = 3.8/(2 \times 32.2)$, or .059 slugs per sec. The thrust produced is $.059 \times 2500$, or 147 lbs. That will give the 7.8-lb. rocket an initial acceleration of about $19g$. The efficiency of propulsion is zero at the start, and rises to a value of $2 \times 1700 \times 2500/(2500^2 + 1700^2)$, or 93% when the final speed is reached.

11-3. Combustion Jet Velocity. When fuels are burned in a combustion chamber, the exit of which is a well-shaped nozzle, the products of combustion

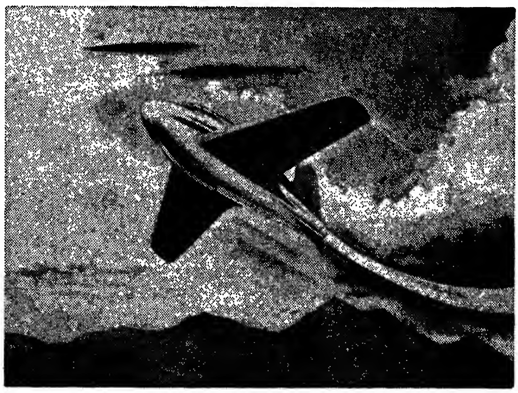


FIG. 11-5. Rocket plane—1945. Limited range—high speed (550 m.p.h.).

attain a high jet velocity since combustion supports a chamber pressure of considerable magnitude. Like any other gas jet, the velocity is a function of combustion chamber pressure. Rockets are burdened with the need of carrying an oxidizing agent as well as a fuel. The product of mass per second of the jet and its velocity is the principal factor in rocket action—the greater this product the more powerful the rocket. The total rocket weight for a shot of definite thrust and duration can be made smaller if the fuel used has a higher jet velocity.

The ideal maximum velocity for a fuel is based on the assumption that all the lower heating value is transferred into kinetic energy of the jet. A pound of hydrogen which yields up some 52,400 B.t.u. per lb. produces 9 lbs. of jet matter when completely burned with pure oxygen. Nine pounds moving at 17,100 ft. per sec. represents kinetic energy equivalent to this heat; consequently that is the ideal maximum velocity of hydrogen. The jet velocities of some other fuels are given in Table 11-1.

The ideal maximum combustion jet velocities of gunpowder are low compared with other fuels. Also the ratio of realized to theoretical jet velocity is lower than with liquid fuels because considerable amounts of fuel are thrown out of the rocket in the form of fiery sparks (carbon). This, along with the better control over rate of combustion afforded by liquid fuels, gives the latter a decided advantage and indicates superiority of the liquid fuel rocket for long-distance rocket shots.

TABLE 11-1. IDEAL COMBUSTION JET VELOCITIES

Fuel	Maximum Exhaust Velocity, ft. per sec.
Acetylene	15,900
Gasoline	14,500
Kerosene	14,500
Alcohol	13,300
Gunpowder	7,000
Smokeless powder	9,600

11-4. Liquid Fuel Rockets. These rockets have been built to use gasoline or alcohol as fuels and liquid oxygen as the oxidizing agent. These liquids are held in separate supply tanks from whence they flow in proper proportions into a combustion chamber. If mixing is adequate, the fuel is almost immediately oxidized, giving up its heat of combustion to the products which thereupon attain a high pressure and temperature. Figure 11-6 illustrates the principle of the liquid fuel "motor." Rocket history of the past two decades is in large part a record of the trials and tribulations of experimenters seeking to perfect a successful liquid fuel rocket motor. Always fighting weight, these pioneers had to overcome problems of the melting and erosion of combustion chambers, proportioning of fuels, fuel feeds, handling methods suitable for liquid oxygen, the stabilization of flight paths, and a host of others. Liquid fuel rockets are still in the experimental stage. In the United States Dr. Robert Goddard led the way in this research with a succession of liquid fuel rockets of progressively better design, his work extending over nearly two decades, 1920 to 1940.

A simplified diagram of the liquid fuel rocket is shown, patterned after some of the early experimental rockets. A combustion chamber forms the head of the rocket. Into this chamber are injected liquid oxygen and gasoline at as steady a rate as possible. Initially ignition is secured from an electrical igniter, after which combustion maintains itself until the fuel or oxygen

supply is exhausted. The rate of fuel feed and the tank capacities determine the magnitude of thrust and duration of the powered ascent. Combustion produces high temperature products. The opening through which these may seek release from the combustion chamber will be made sufficiently small so

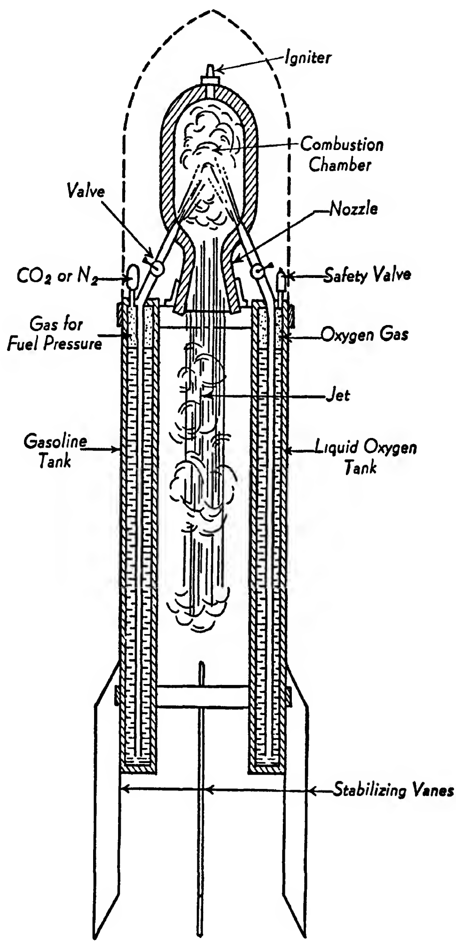


FIG. 11-6. Early type liquid fuel rocket.

that the products of combustion must build up a considerable fluid pressure before they flow out through the nozzle as rapidly as they are formed. This expansion to atmospheric pressure produces a jet of high velocity which will, however, fail by a large margin of attaining ideal maximum jet velocity. The reasons for this failure are:

1. The oxygen and fuel are not injected in the ideally correct proportions, or else are not sufficiently mixed before or during combustion.

2. Combustion is not completed before the gas enters the nozzle because of the
   a. Time factor, i.e., small chamber and high rate of combustion.
   b. Dissociation factor, i.e., incomplete combustion within the chamber because of the thermal equilibrium of a percentage of fuel and oxygen possible at elevated temperatures.
3. Expansion ratio is not great enough to lower the internal energy of the exit gases to the initial state.

With regard to item 2b above, the dissociation action takes some time for completion—which is not ordinarily allowed to it in the rocket combustion chamber and nozzle. Therefore high chamber temperatures and nearly adiabatic expansion may be caused to prevail in rocket combustion.

During expansion the temperature falls and the dissociated products may eventually all recombine, liberating the full potential heating value. But since some of the combustion is thus accomplished during the expansion, perfect adiabatic expansion is destroyed, and the nozzle fails to convert the ideal available heat into kinetic energy. For this reason, calculations based on equations premised on adiabatic expansion will be in some error for rockets, but not as much so as for spark ignition engines. The expansion through gas turbine and jet engine nozzles is not subject to the same consideration because the use of air rather than oxygen, and in considerable excess too, usually insures that the temperatures are low enough (1000° F to 1500° F) to avoid any substantial amount of dissociation.

The rocket in Figure 11–6 is poorly shaped for high velocity flight. It represents a type in use when the possibility of successful liquid fuel rocket flight of any sort was in doubt. The trailing tanks and fins gave it a skyrocket type of stability. For several reasons, rear end position of the motor is preferable, particularly in high-speed flight. The necessary rocket contents, fuel, controls, etc., can be located in a long sleek cylindrical case, sharply pointed at the nose to help penetrate the air at sonic speeds. Stability in flight has been secured by the use of stabilizing fins and gyro-operated control vanes. Examination of the German liquid fuel rocket (V-2) will convey some idea of the progress that has already been made in this field.

As the illustration on page 292 shows, the rocket body is cylindrical and elongated. The sharply pointed nose consists of a one-ton explosive warhead back of which are located the radio control equipment and the gyroscopic stability devices used for remote control of the flight. But the interior is mainly filled by tanks containing the liquid propellents. The after part of the rocket body is occupied by the combustion equipment. A turbine is used to drive rotary pumps which withdraw the alcohol and liquid oxygen from their tanks and force them into the combustion chamber. The efflux of prod-

ucts of combustion from this chamber takes place through a nozzle about 18 in. in diameter. What the combustion chamber pressure might be has not been publicly stated. It has been estimated that the jet velocity was about

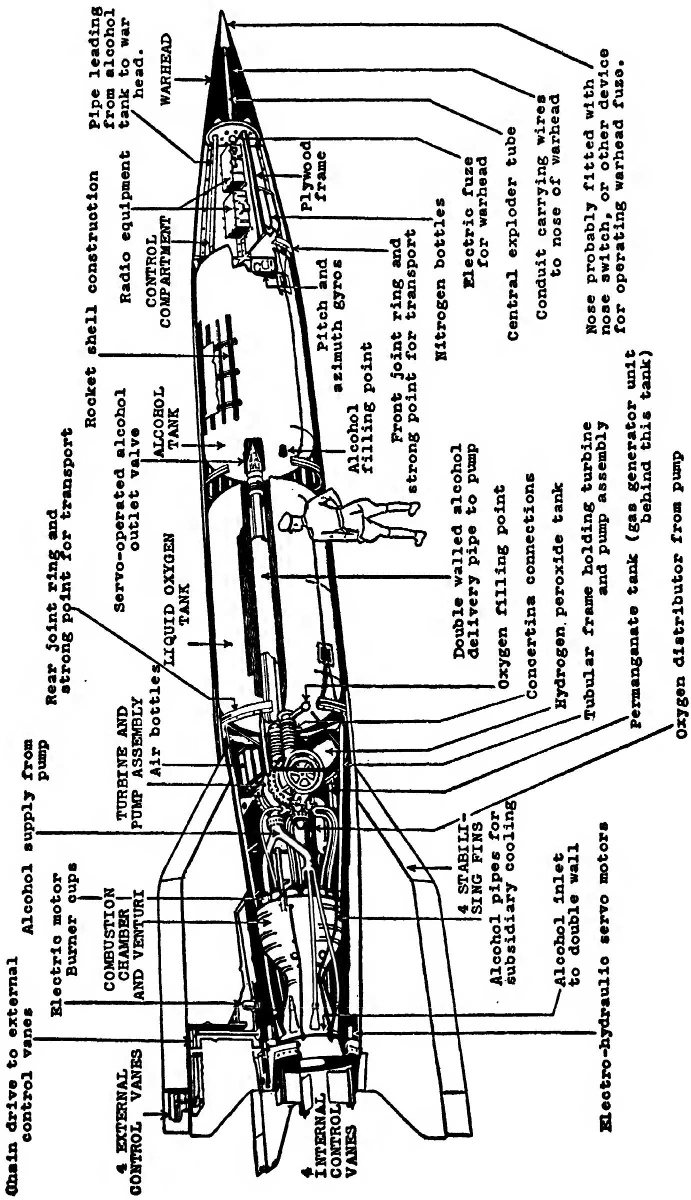


FIG. 11-7. Liquid fuel rocket—1945. Sectional drawing shows the German V2 rocket bomb. (Courtesy British Information Service.)

6000 ft. per sec., and this, coupled with the fact that some 8.5 tons of propellants were injected into a hemispherical chamber about 4 ft. in diameter in a little over a minute, implies that the combustion chamber pressure was of a

high order. From its pump the fuel is led through jackets surrounding the nozzle and combustion chamber to cool them enough to prevent their destruction. As the heat thus absorbed is retained by the fuel on its way to the burners, this system is called regeneration, and the rocket motor designated a *regenerative motor*. The warm alcohol is mixed with liquid oxygen in some eighteen burner cups attached to the forward surface of the combustion chamber. The operation of the rocket is supposed to be about as follows. The fuel and oxygen tanks are filled, their relative capacities corresponding closely to those required for the ideal combustion of methyl alcohol with oxygen. Compressed air from small bottles then forces calcium permanganate into hydrogen peroxide, accompanied by an exothermic chemical reaction which results in the formation of superheated steam. The steam, of course, was produced solely for powering the pump turbine. The pumps transfer the oxygen directly to the burners and the alcohol to them via the cooling jackets. Ignition is initiated electrically, and thereafter maintains itself as long as the fuel holds out and the turbine continues to turn. The rocket takes off as soon as the jet builds up a reaction thrust equal to its weight. Were the 8.5 tons burned in 70 sec., the exhaust would consist of 7.5 slugs per sec., which, moving at 6000 ft. per sec., would induce a reaction thrust of $(6000 \times 7.5)/2000$, or about 22.5 tons. The rocket is said to weigh 12 tons ready for launching; consequently a net 10.5 tons is available for acceleration. This would provide an initial acceleration of a little less than $1g$. As the fuel is consumed the rocket mass is diminished and the available thrust can propel the rocket at greatly increased acceleration. The combustion is said to propel this rocket about 25 miles high, after which its momentum carries it upward another 40 miles.

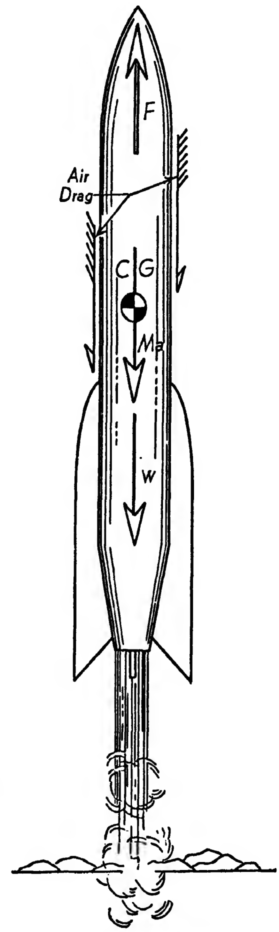


FIG. 11–8. Forces on the rocket.

The forces acting on a liquid fuel rocket aimed for vertical ascent are shown in Figure 11–8. The exhaust jet produces a reaction of F lbs., which, of course, must equal or exceed the sum of air drag D, and dead weight w. When F exceeds $D + w$, then $F - (D + w)$ is an accelerating force that can increase the velocity, u, of the rocket. An equation balancing the net thrust against inertia yields, for vertical rocket acceleration, $a = (g/w)[F - (D + w)]$

in which g is the acceleration of gravity. Supposed variations of some of these quantities with extreme altitude are shown by the accompanying graph. Rocket flights to a few miles of height above the earth's surface are today's wonder. Ascents to such heights that the gravitational field is noticeably weaker will probably come some day. Several books have been written which seriously weigh the pro's and con's of rocket flight into interplanetary space.

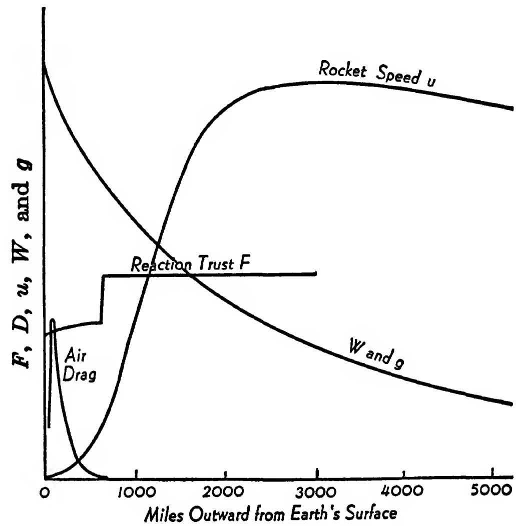


FIG. 11–9. Supposed variation of factors in rocket repulsion. (Low initial combustion rate to keep speed and air drag limited until the atmosphere is left behind.)

From a former state of pessimism following analyses of requirements to be met in attempting a rocket-to-the-moon, there now appears a trend of thoughtful reasoning which admits the possibility.

Example 1: Liquid oxygen and gasoline are to be burned in a rocket combustion chamber rapidly enough to accelerate the rocket so that it will be travelling a mile a minute after the first 100 ft. of travel. Other data are: Magnitude of the rocket is set by working for 35,000-lb. thrust, for which size it is thought the rocket body will weigh about 30% as much as the propellants. Jet velocity 9150 ft. per sec. Air drag neglected.* Flight path vertical. It is desired to determine (a) the capacity of the propellent tanks, (b) the length of time during which the thrust acts (burning time).

If acceleration is assumed uniform during 100 ft. of travel at the end of which u = 88 ft. per sec., the acceleration can be computed from that well-known law of uniformly accelerated motion, $a = \dfrac{u_1^2 - u_0^2}{2d}$. $u_0 = 0$, $u_1 = 88$ ft. per sec. $d = 100$ ft., hence $a = 38.8$ ft. per sec. Assuming mass constant during the first 100 ft., the mass that can be accelerated to a mile a minute is found from $F = Ma + Mg$, in which the second term is the dead weight which must be overcome before acceleration can begin. So, $35{,}000 = M(38.8 + 32.2)$, whence $M = 493$ slugs. Actually this is

* This does not mean that it is negligible. This assumption is made here to simplify the problem, but air drag would be considered in the calculations for real rocket flight.

the average mass for the first 100 ft. A preliminary trial showing about 2% of the fuel consumed during this period, M_1 is assumed to be 1.01 × 493, or 498 slugs. The rocket body mass $M_2 = 30\% \ (M_1 - M_2)$. $\therefore M_2 = 115$ slugs. Propellants = 498 − 115 = 383 slugs.

(a) 383 slugs represent 12,340 lbs. of products. To find the quantities of O_2 and gasoline involved, consider the latter to be represented by C_8H_{18} (Octane). Assuming that propellants will be combined in ideal proportions, a combustion reaction can be written after the pattern shown on page 66.

$$1 \text{ lb. } C_8H_{18} + 3.5 \text{ lb. } O_2 \rightarrow 4.5 \text{ lbs. of products,}$$

or 1 lb. of fuel produces 0.14 slugs of products. Evidently 383/0.14 = 2735 lbs. gasoline and 3.5 × 2735 = 9570 lbs. liquid oxygen are to be carried.

(b) The rate at which propellants must be ejected to obtain 35,000 lbs. of thrust is found from the equation for rocket thrust,

$$F = mv.$$

So

$$35{,}000 = m \times 9150.$$

$$m = 3.83 \text{ slugs per sec.}$$

Assuming uniform rate of burning, the time consumed is 383/3.83 = 100 sec.

Example 2: Estimate how high the rocket of Example 1 would ascend in the absence of air drag.

Although thrust remains constant, mass decreases and acceleration increases. If advanced mathematics are to be avoided, this problem will require the total time to be broken up into increments and the rocket mass assumed constant over the increment. The following table uses 10 sec. increments, during each of which 38.3 slugs are expelled.

TABLE 11–2. COMPUTATION OF ROCKET ASCENT

t sec.	M slugs	M_{av} slugs	Net a_{av} ft./sec.^{2}	u ft./sec.	u_{av} ft./sec.	Δh_p ft.	h_p ft.	t sec.
0	498.0			0			0	0
		479.8	40.8		204	2,040		
10	459.7			408			2,040	10
		440.7	47.3		645	6,450		
20	421.4			881			8,490	20
		402.3	54.8		1,155	11,550		
30	383.1			1,429			20,040	30
		364.0	64.0		1,749	17,490		
40	344.8			2,069			37,530	40
		325.7	75.3		2,446	24,460		
50	306.5			2,822			61,990	50
		287.4	89.5		3,270	32,700		
60	268.2			3,717			94,690	60
		249.0	118.3		4,308	43,080		
70	229.9			4,900			137,770	70
		210.8	134.0		5,570	55,700		
80	191.6			6,240			193,470	80
		172.4	170.8		7,094	70,940		
90	153.3			7,948			264,410	90
		134.2	228.6		9,091	90,910		
100	115.0			10,234			355,320	100

The propulsion efficiency rises from zero to become 100% sometime during the last 10 sec. of combustion. At the end of powered ascent the propulsion efficiency is 99.3%. The value of u obtained from the equation $u/v = \log_e (M_1/M_2)$ is higher than the table yields because the latter takes into account the gravitational field. This field was assumed to have an acceleration of 32.2 ft. per sec. At the altitude of 355,320 ft. this field is noticeably weaker; so rather than use the simple law of freely falling bodies ($u^2 = 2 \times 32.2 \times d$) for estimation of height to which the rocket will continue under its momentum, the following equation, which takes into account the variation of gravity, is used.

$$h_m = \frac{K_1 h_p + K_2 u^2 (K_2 + h_p)}{K_1 - u^2(K_2 - h_p)},$$

in which h_m = Maximum altitude of the rocket, above the earth's surface, ft.
h_p = Altitude gained under powered flight, ft.
u = Velocity (vertical) gained under powered flight, ft. per sec.
$K_1 = 29 \times 10^{15}$; $K_2 = 2.12 \times 10^7$.

Substituting 10,234 from the table for u, and 355,320 for h_p, the above equation yields 2,165,000 ft. (410 miles) height reached by this rocket.

11–5. Jet Engine. Technically this might include any device for propulsion which uses the reaction force obtained from the acceleration of a fluid stream. However, it is proposed to restrict the present treatment to combustion jets, wherein the reaction of a powerful jet of products of combustion expelled rearward from the engine is the source of propulsion. Present-day jet engines meeting this specification are classified as follows:

1. Gas turbine systems.
2. Resonance ducts.
3. Aero-thermodynamic ducts (Athodyd).

All of these draw in air for combustion from the surrounding atmosphere, and it is in this respect that they exhibit their principal difference from rocket motors.

Propulsion by jet action never becomes efficient unless the propelled body is moving at high speed. Airplanes furnish this condition, in some measure, as do also the so-called "flying bombs." In general, surface vehicles do not. Jet engine development therefore was nurtured in the aviation field, receiving great impetus from wartime demands for planes with lighter, more powerful engines, especially those well adapted to the high-speed range.

11–6. Gas Turbine Systems. One method of producing jet reaction is to impart a high velocity to the fluid via adiabatic expansion. But this necessitates a pressure drop. Since the discharge is back to the same atmosphere from whence the air was initially drawn, a means of compression has to be provided. Briefly, this system has a gas-turbine-driven compressor. Both centrifugal and turbine type (axial flow) compressors have been used. Between discharge from the compressor and entrance to the turbine, the air re-

ceives heat by the direct combustion in it of a suitable quantity of fuel. This increases the volume and temperature at constant pressure. The energy level is then sufficiently high so that the work of compression can be taken out of the gases by the turbine and have remaining a large quantity for imparting the high final kinetic energy to the jet.

The diagram, Figure 11–10, shows the basic features of the turbo-jet engine. The single rotating shaft mounts the impeller of a centrifugal compressor and the wheel of a single-stage impulse gas turbine. Air, sometimes at subzero temperatures, is rammed into the engine cowling by the forward

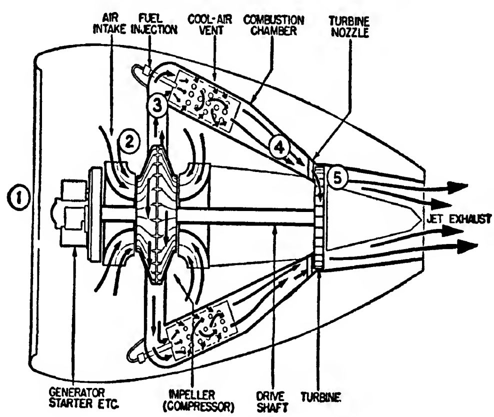


FIG. 11–10. Principle of the combustion gas turbine jet engine. (Courtesy General Electric Co.)

motion of the whole engine. It speedily finds the air intake of the compressor and quickly is discharged in a compressed condition into the combustion chamber. Here fuel (kerosene) is burned in it in sufficient quantity to raise its temperature to incandescence, say 1500° F at the turbine nozzle. To obtain as complete combustion as possible, and to cool the chamber walls, the air-fuel ratio is temporarily reduced by feeding only part of the air through the burner. The remainder enters the combustion chamber through cooling vents. After partial expansion in the turbine nozzles the gas flows at high speed through the turbine blading, spinning the wheel at many thousand revolutions per minute. All the turbine output is used to drive the compressor; consequently only a portion of the available energy is consumed in this way. The gases emerge from the turbine wheel with considerable velocity and some positive pressure above final exhaust pressure. This is employed by the exhaust nozzle to raise the gases to the final jet speed by adiabatic expansion. It is seen that the compression and combustion process is devoted to readying a flow of gas, composed of products of combustion and excess air, for adiabatic expansion through a nozzle. The gas turbine is merely incidental, a means of compressor drive.

Theoretical considerations of energy and of momentum yield, as we have seen, equations for thrust and efficiency based on engine speed u, jet speed v, and mass flow m. This approach has the virtue of simplicity but fails to include the factors which are responsible for whatever numerical value v may possess. With reference once again to Figure 11–10, note that five stations along the line of flow of the fluid are selected as state points in a description of the engine process from the thermodynamic viewpoint. State 1 is the free atmosphere at temperature T_1. The ram of air into the engine cowl is of the nature of a reversed and imperfect expansion of a gas jet. Borrowing an equation from page 89, with suitable subscripts for the case under consideration, the ideal ram pressure $P_2 = P_1 \sqrt[z]{\frac{u^2 z}{64RT_1} + 1}$. The relation $T_2/T_1 = (P_2/P_1)^z$, whose use was necessary in formation of the equation for P_2 from the gas jet equation, will also serve to determine the compressor intake temperature, for the actual ram pressure P_2 will be 80% to 90% of the ideal. Use symbols as follows:

$$W_c = \text{Ideal compressor work} \left\{ = \frac{RT_2}{z}\left[\left(\frac{P_3}{P_2}\right)^z - 1\right]\right\}.$$

η_c, η_e = Internal efficiencies of compressor, turbine.
R = 53.4 for air, nearly the same for gas turbine products.
$z = (c_p - c_v)/c_p$. [also $z = (\gamma - 1)/\gamma$].
c_p, c_v = Specific heats, constant pressure and constant volume.
h = Enthalpy.
J = Mechanical equivalent of heat.

The remaining steps in determining the magnitude of the final velocity, which call v, for a limiting T_4, are based on a pound of air.

$$h_3 - h_2 = \frac{W_c}{J} = c_p(T_3 - T_2).$$

This defines the temperature T_3 to be expected from a pressure ratio P_3/P_2. The combustion process which raises the temperature from T_3 to T_4 is essentially constant pressure, making the necessary heat addition $c_p(T_4 - T_3)$, although some adjustment might be made for a drop of pressure between points 3 and 4. Since all the turbine output goes to the compressor, the enthalpy at point 5 can be determined by $h_4 - h_5 = W_c/J\eta_c +$ energy required by the auxiliaries. Considering $T_{5'}$ the temperature at exit from turbine nozzle, and T_5 the temperature at exit from turbine blades,

$$h_4 - h_{5'} = \left(\frac{W_c}{J\eta_c} + \text{auxiliary energy}\right) \times \frac{1}{\eta_e} = T_4 c_p - T_{5'} c_p.$$

Thus $T_{5'}$ is fixed, and also the pressure $P_{5'}(= P_5)$ from the polytropic relation $(P_4/P_{5'})^z = T_4/T_{5'}$. The actual temperature drop from 4 to 5 is $\eta_e(T_4 - T_{5'})$. With P_5 and T_5 the final jet velocity is given by the jet equation

$$v = 8\sqrt{\frac{RT_5}{z}\left[1 - \left(\frac{P_1}{P_5}\right)^z\right]}.$$

Example: Calculate the propulsive efficiency and overall thermal efficiency (assuming complete combustion) of a jet engine moving at 400 mi. per hr. Data given: P_1 7.8 psi. Pressure ratio $P_3/P_2 = 4$, temperature, t_1 0° F, $t_4 = 1500°$ F, $\eta_c = 85\%$, $\eta_e = 80\%$, energy for auxiliaries 2%, other data assumed as needed during the solution.

$$\text{Ideal ram pressure } P_2 = 7.8\sqrt[z]{\frac{(588)^2 z}{64 \times 53.4 \times 460} + 1} = 9.65 \text{ psi.}$$

if z is taken as .286. The ram is therefore 9.65 − 7.8 = 1.85 psi. Probably only 90% of this will be realized, making P_2 9.47 psi. Assuming adiabatic compression to 9.47 psi.,

$$T_2 = T_1\left(\frac{9.47}{7.8}\right)^z = 486° \text{ (26° F).}$$

$$\text{Ideal compression work } W_c = \frac{53.4 \times 486}{z}[4^z - 1] = 44{,}200 \text{ ft. lb./lb. air.}$$

$$h_3 - h_2 = \frac{44{,}200}{778 \times .85} = c_p(T_3 - 486).$$

Taking $c_p = .24$,

$$T_3 = 765° \text{ (305° F).}$$

The heat input, Q, in the combustion chamber is $c_p(T_4 - T_3)$, for which c_p is estimated (handbook sources) to be 0.28. $Q = .28(1960 - 765) = 335$ B.t.u./lb. air.

$$P_4 = P_3 = 4P_2 = 37.9 \text{ psi.}$$

$$h_4 - h_{5'} = \frac{1}{.80}\left[\frac{44{,}200}{778 \times .85} \times 102\%\right] = .28(1960 - T_{5'}).$$

$$T_{5'} = 1656° \text{ R.}$$

$$\frac{P_4}{P_{5'}} = \sqrt[.26]{\frac{1960}{1656}}.$$

Here z is reduced to 0.26 because of the high temperatures at this point.

$$P_5'(= P_5) = 19.8 \text{ psi.}$$

$$T_5 = T_4 - .80(1960 - 1656) = 1717° \text{ R.}$$

Final jet velocity, based on nozzle correctly shaped for expansion to P_1, a z of .27, and a nozzle velocity coefficient of 98%, is

$$v = .98 \times 8\sqrt{\frac{53.8 \times 1717}{.27}\left[1 - \left(\frac{7.8}{19.8}\right)^{.27}\right]} = 2160 \text{ ft. per sec.}$$

The propulsion efficiency is $2(uv - u^2)/(v^2 - u^2)$, so

$$\eta = \frac{2(588 \times 2160 - 588^2)}{2160^2 - 588^2} = 42.7\%.$$

The thrust, F, should be $(1/32.2)(2160 - 588)$ lbs. per lb. air, and the overall thermal efficiency $\frac{\text{Thrust work}}{778Q}$.

$$F = 48.8 \text{ lbs. per lb. air flowing per sec.}$$

$$Fut = 48.8 \times 588 = 28{,}700 \text{ ft. lbs. per lb. air per sec.}$$

$$\text{Thermal efficiency} = \frac{28{,}700}{778 \times 335} = 11\%.$$

To obtain good thermal efficiency the temperature T_4 needs to be as high as conditions permit. The cooling of these engines is therefore a critical feature of their design. Cooling air bled off the compressor is used to air-jacket

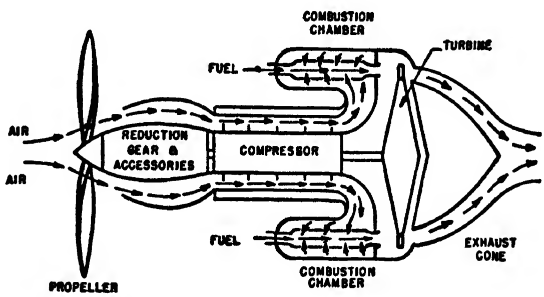


Fig. 11–11. G-E propeller drive gas turbine. (Courtesy General Electric Co.)

many of the high-temperature regions. Air circulation through hollow turbine blades and other ingenious features have characterized these designs. Although not impressive in thermal efficiency, these pure jet engines are capable of powering airplanes of higher speed than the propeller type engines whose propeller blades are the first part of the airplane to be affected by atmospheric compressibility.

A new type of power plant has been developed, having both propeller and jet output. Designed primarily to drive large high-speed military transports and bombers, this gas turbine with propeller is designed for installation in the wings of multi-engined aircraft or in the nose of a single-engine plane. A diagram of it shows that the air rams into the nose of the engine through ducts opening forward. This air is compressed by axial flow units in the forward part of the engine and then forced into combustion chambers. There fuel is injected and burns intensely. This raises the temperature and velocity of the gases, which then, with great energy, strike the buckets of the turbine wheel.

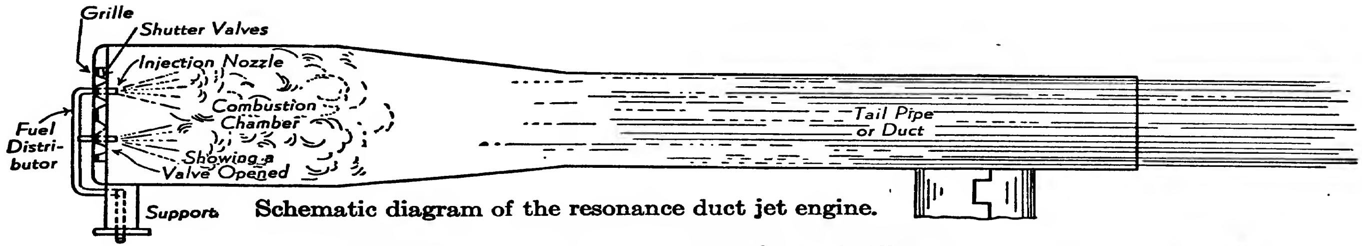
The turbine, spinning more than 10,000 times a minute at a temperature over 1500° F, absorbs the *major part* of the energy in the gases.

The turbine powers the compressor and through reduction gears drives the propeller. Reactive thrust created by the energy remaining in the gases passing through the turbine wheel and, discharging rearward, is utilized in jet propulsion.

The range of planes powered by gas turbines of this type will be extensive. When flying at slow speeds, at low altitudes, the gas turbine uses more fuel than a reciprocating engine would use cruising under similar conditions, but when operating at full power the turbine uses less fuel than a conventional engine operating at maximum power. The turbine functions most efficiently at high altitudes where the air is colder than at lower altitudes.

No complicated ignition or carburetion or timed injection system is needed on the combustion jet engine, but starting is more of a task as the compressor must be set spinning at high speed in order to initiate the cycle. Powerful electric starting motors or auxiliary S.I. engines are used.

11–7. Resonance Duct Engine. In 1944 a new type of jet engine made its appearance in considerable numbers on a war weapon. Without touching on its value as a weapon, the technical features of its propulsion system are described. The "buzz bomb," or V-1, to mention two of the names ascribed to this flying bomb, was an unmanned aircraft carrying an explosive charge which was detonated when the bomb crashed at the end of its one to two hundred mile flight. Gyro stabilizing, and other controls which could be preset, guided it to the target. By far the most interesting thing about this weapon was the jet engine that propelled it. Although possessing poor thermal efficiency, it was extremely light weight, simple, and inexpensive to construct, just the features needed for the expendable nature of its use. The principle of the "intermittent duct engine" is explained by Figure 11–12. The duct itself is a long tube (11 ft.) enlarged at the forward end to serve for the combustion chamber. Across the front is a grill equipped with large numbers of thin spring steel flap valves which will open inward to allow air to enter, but will close when fuel burning in that air raises the combustion chamber pressure. Following each explosion a sound wave and column of gases travel down the duct and a partial vacuum is induced behind the grill. So more air enters, mixes with fuel that is sprayed into the chamber continuously, and explodes. These explosions follow each other in rapid sequence. Rates of 40 cycles per sec. have been mentioned. The ignition is by electric spark. However, after the combustion chamber is hot, ignition is continued by residual flame. The duct must be of the correct length to harmonize the explosion impulse waves with the natural period of oscillation of the grill valves. The source of thrust is, of course, the change in momentum of the gases shot rearward out of the duct.



Schematic diagram of the resonance duct jet engine.

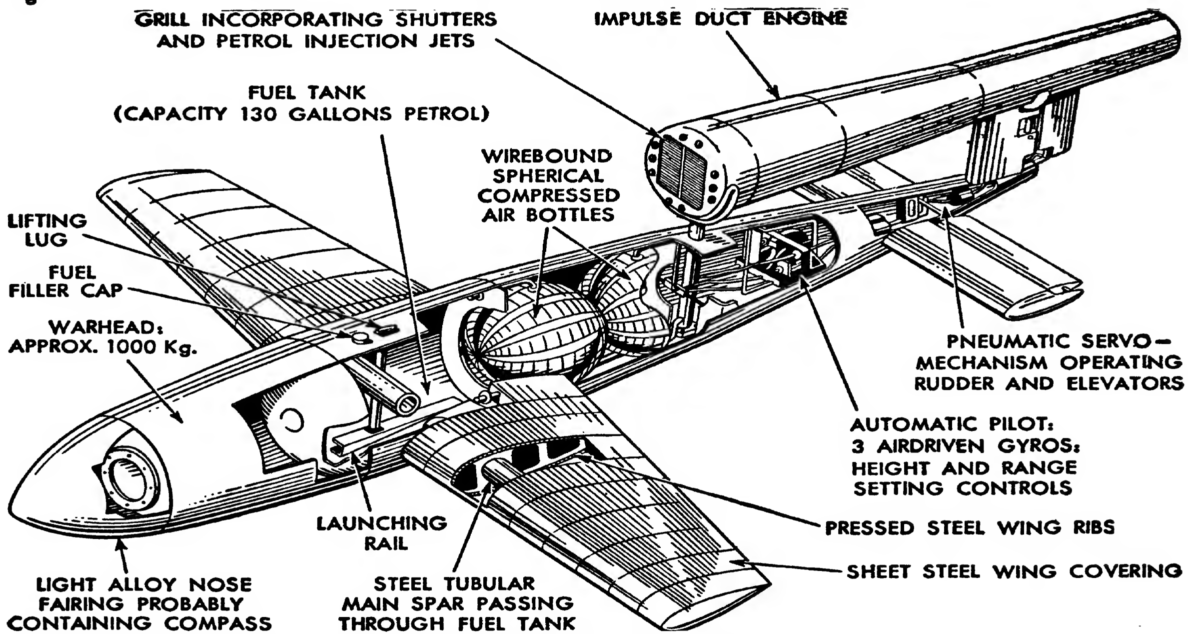


FIG. 11-12. Flying bomb with jet propulsion. Resonance duct type. (Courtesy British Information Service.)

The flying bomb, which was able to make something like 300 mi. per hr., is shown in an accompanying figure. Because of the low thermal efficiency, it needed to have aboard at the start of the flight about one gallon of gasoline for each mile to be covered. This was carried in a fuel tank from which it was forced, by compressed air pressure, into fuel jets located in the front grill. A big disadvantage of this engine, and one which ultimately prevented its becoming a decisive weapon, was the trouble of getting it started. It had to be projected at about 250 mi. per hr. from a long launching track before the resonance action could begin to function. This rendered the launching sites large vulnerable targets for enemy counter action. Some of these flying bombs were carried aloft by bombers and launched in flight. Although they were effective only against large targets, in another war, when equipped with television transmission in the nose, and perfected for longer ranges, they might offer a major and serious offensive threat.

11–8. Athodyd. This simple continuous jet engine is known by few, and understood by fewer. It is known to have been the subject of considerable wartime research, but the results, if any, have not appeared prominently as yet. Not that the *aero-thermodynamic-duct* principle is new. On the contrary, it has been a subject for thought and debate for at least three decades. But its field of usefulness seems to lie in speed ranges only now being approached in aviation.

The device is a nozzle set into a well streamlined body containing fuel tanks and a system for injecting the fuel into the combustion region. It is not self-starting. If fuel is injected and ignited with the duct stationary, the exhaust will blow out of the duct in both directions, for there is nothing there to cause it to do differently. But when the duct is in motion at high speed, the air ram into the front end will block the possibility of exhaust there unless it is attempted to burn the fuel too rapidly. The thrust is theoretically

$$F = \frac{\rho u^2}{2}(A_m - A_e),$$

in which ρ = Mass density of the air.
u = Duct speed.
A_m = Mouth area.
A_e = Entrance area.

The $(A_m - A_e)$ term must be limited in magnitude because of certain thermodynamic and aerodynamic considerations. Then large thrusts must arise out of high speeds which enlarge the magnitude of dynamic pressure $\rho u^2/2$. Power is proportional to the product of thrust and velocity, which makes it proportional to u^3. Thus we find Athodyd to be a device suitable only for high speeds, but its power, insignificant at conventional speeds, rises mightily

as those high speeds (thought to be near the speed of sound) are approached. The ideal efficiency is 100%, but whether that can be approached in an actual case depends on whether the duct design permits a decent amount of combustion to occur before the jet carries out of the duct.

PROBLEMS

1. A rocket burns a fuel with resulting jet speed of 1000 ft. per sec. Plot the instantaneous efficiency of propulsion of this rocket against its speed for a range from zero to 2000 ft. per sec. rocket speed. 1 in. = 20%, 1 in. = 400 ft. per sec.

2. In a certain jet engine the exhaust velocity is 1500 ft. per sec. while the forward engine speed is 450 mi. per hr. What is the efficiency of propulsion?

3. What "mass ratio" is necessary in order to secure the velocity of liberation from the earth in a rocket burning gasoline in liquid oxygen to the ideal combustion jet velocity?

4. Repeat Problem 3 for a rocket burning hydrogen in oxygen.

5. A powder rocket is shot horizontally with an exhaust velocity averaging 2000 ft. per sec. Its initial weight was 2.5 lbs., of which 1.0 lb. was powder. Neglecting air drag, estimate the maximum velocity attained by this rocket.

6. The powder in the rocket of the preceding problem burns at the rate of $\frac{1}{4}$ lb. per sec. Calculate the velocities attained each quarter second of the first second of flight. Estimate the distance traversed by use of the average velocity during each quarter-second interval.

7. Assume that smokeless powder rockets are developed to where the exhaust velocities are half the theoretical value. Find the weight of powder needed in a 4-lb. (empty) rocket in order to give it a maximum horizontal velocity of 1850 ft. per sec. Neglect air resistance. How fast must the powder burn (lbs. per sec.) for the rocket to have an acceleration of $15g$ when the fuel is half consumed?

8. The exhaust velocity of a gunpowder rocket is 1800 ft. per sec. Its weight is 15 lbs., including 8 lbs. of powder. It achieves its maximum velocity 3 sec. after ignition. With horizontal launching, what is its maximum velocity? Assuming that the fuel burned uniformly, what is the acceleration at the end of each of the first 3 sec.?

9. Methyl alcohol, CH_4O, has a heating value of 10,270 B.t.u. per lb., while ethyl alcohol, C_2H_6O, has one of 13,170 B.t.u. per lb. Both these are *higher* heating values. Which alcohol is represented in Table 11-1? Note: Only the lower heating value can produce kinetic energy. It will be necessary to write a combustion reaction for one pound of alcohol with oxygen, following the method described in Chapter 3, in order to obtain (1) the mass of the products, (2) data for obtaining lower heating value.

10. Check the ideal combustion jet velocity, of gasoline, C_8H_{18}, as given in Table 11-1. The lower heating value is 19,250 B.t.u. per lb. See note offered in Problem 9.

11. A liquid fuel rocket using gasoline (C_8H_{18} which ideally needs 3.5 lbs. oxygen per lb.) fuel is to produce an exhaust jet of 5 slugs per sec. for a full minute. What capacity fuel and oxygen tanks are needed?

12. A rocket like Figure 11–7, with diameter of 30 in., has 15 ft. of length available for propellent tanks. How many seconds can combustion be maintained at the rate of 5 slugs per sec.? Propellants are gasoline (C_8H_{18}, specific gravity .76), and liquid oxygen (O_2, specific gravity 1.3).

13. A liquid fuel rocket is to produce a steady thrust of 40,000 lbs. for 80 sec. Exhaust velocity 9000 ft. per sec. The rocket body will weigh a quarter as much as the propellants. What is the "mass ratio," and the initial weight (lbs.)?

14. If the answer to Problem 13 were 7.2 tons, to what height could the rocket ascend in the absence of air drag?

15. Repeat Problem 13 for an exhaust velocity of 10,000 ft. per sec. and a duration of 100 sec.

16. If the answer to Problem 15 were 8.0 tons, to what height could the rocket ascend in the absence of air drag?

17. Sketch the jet engine of Figure 11–10 and label it appropriately with the data given or calculated in the sample example, page 299. Illustration should occupy approximately 8 in. × 11 in. of space.

18. The jet engine of an airplane flying at 30,000 ft. altitude with a speed of 500 mi. per hr., receives air ram with an efficiency of 90%. Estimate the air temperature at the compressor intake.

19. A certain airplane has an engine similar to the one shown in Figure 11–10. When flying at 500 mi. per hr. at 25,000 ft. altitude, what is the air temperature, ideally, inside the forward end of the cowling?

20. A jet engine operates on a pressure ratio of 4.5. With a maximum allowable combustion chamber temperature of 1350° F, how many B.t.u. will be added to each pound of air by combustion? Take compressor intake condition of 10 psi., 30° F, compressor efficiency 80%. Take same c_p's and z as in example, page 299.

21. Data of Problem 20. How many B.t.u. would have been added had the maximum temperature been 1500° and the compressor efficiency 82%?

22. Given a jet engine with combustion chamber temperature of 1500°, pressure 45 psi. abs., turbine engine efficiency 85%, necessary turbine output, per pound air flowing, of 50,000 ft. lbs., calculate the thrust produced per pound of air flowing per second. Atmospheric pressure 10 psi., $c_p = .28$, $z = .26$. Airplane speed 400 mi. per hr. Exhaust nozzle velocity coefficient 95%.

23. State 4 (Figure 11–10) is 50 psi., 1300° F, state 1 is 4 psi. η_e 80%, compressor input 43,000 ft. lbs. per lb. air flowing. Auxiliaries consume 1000 ft. lbs., c_p .28, z .265. Exhaust nozzle velocity loss 5%. Airplane speed 450 mi. per hr. Calculate the propulsive efficiency.

24. The exhaust jet velocity from a certain jet engine is 1800 ft. per sec. at an airplane speed of 420 mi. per hr. The turbine output per pound air flowing is 52,000 ft. lbs. Find the necessary air flow per second and the horsepower of the turbine for an engine thrust of 5000 lbs.

25. An Athodyd moving at 600 mi. per hr. in air whose density is 60% of sea-level value has an internal duct 6 in. in diameter at the front end, 10 in. at the rear. What thrust, in pounds, should it develop?

CHAPTER 12

External Combustion Power Plants

12-1. External Combustion. External combustion is action in contrast to internal combustion where products of combustion form the working fluid. In external combustion the fuel is burned, and its heat is developed, *outside* the "prime mover." So a furnace is provided where fuel can be properly burned with air. The heat of combustion is then transferred to a *working medium* * which carries it to the prime mover—an engine or turbine. There the fluid medium will be caused to undergo a process of expansion which will convert a considerable portion of the heat energy into mechanical work, thus accomplishing the purpose for which the heat power plant is called into existence.

The working medium is usually water, although other fluids have occasionally been employed. Hence "steam power plant" is practically synonymous with "external combustion system."

External combustion systems use a vaporizable fluid because through evaporation much more heat can be "loaded on" a unit quantity of the working medium than would be the case if only sensible heat changes were employed, as in the gas turbine. Furthermore, vaporization, and its companion process condensation, are physical changes and are simpler to handle and control than chemical changes. External combustion systems *could* be operated on some other basis than evaporation and condensation. It might be possible to use chemical decomposition and recomposition instead of evaporation and condensation. However, the advantages of using a stable vapor subjected only to physical changes have prevailed and it has become the common method of transformation of heat into work in these external combustion cycles.

The external combustion power plant is summarized thus. Potential heating value of a fuel is realized as heat energy by combustion, which is made to occur in a furnace. Most of this heat is transferred to a working medium in a *heat absorber.* This will actually be a boiler, so-called because in it the working fluid is converted from the liquid to the vapor phase by "boiling" action. The fluid having absorbed whatever heat is necessary to convert it to a vapor of suitable state is led through pipes to the prime mover, which might be

* The gas turbine plant has "external combustion"; however, products of combustion form the working medium.

called a *heat utilizer*. Whether it be engine or turbine, it relieves the vapor of some of its heat,* via adiabatic expansion, causing partial condensation. Mechanical work appears at the prime mover shaft and low-pressure waste (exhaust) steam is discharged. This describes the simple steam power plant. It does not tell about salvaging some of the heat in the exhaust steam, nor does it mention how the fluid originally found its way into the boiler. It does show that the primary elements of such a system will be (1) a combined heat generator and absorber, (2) a prime mover, and (3) a working medium to carry heat from (1) to (2).

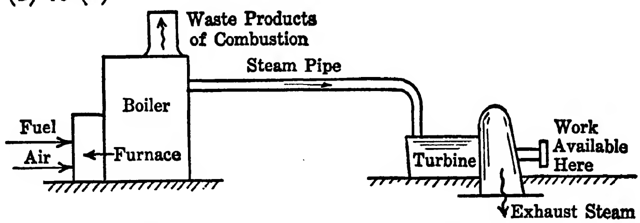


FIG. 12–1. Elements of the external combustion power plant.

12–2. Vapors for Heat Power Plants. There are several fluids whose physical characteristics allow their use as a heat power working medium. Among them are water, sulfur dioxide, mercury, and a few artificial compounds. A good substance for this purpose should not be poisonous, or corrosive to iron and steel. Its range of thermal properties should be suitable for the service. A capacity to hold heat energy at a high temperature without developing at the same time a high fluid pressure is quite desirable. High pressures increase equipment costs, but are often countenanced in order to secure the advantages of attendant high temperature. On the whole, water is a fairly satisfactory working medium, although other substances may better it in some specific property. Furthermore, water is nearly universally prevalent. It is cheap, abundant, and relatively stable. Its latent heat of vaporization is high, and its volume in the liquid phase is quite small compared to its vapor phase volume at usual pressures. These favorable characteristics have secured for the water substance a dominating position in the vapor cycle field. To a very limited extent mercury has been employed because its vapor pressure remains moderate even at very high temperatures.† Its use is restricted to the high temperature portion of the heat → work conversion, steam being employed to complete the action.

12–3. The Steam Power Plant. The conversion of heat into work in the external combustion system is usually worked out in several separated pieces of equipment, connected by piping. The whole assembly is called a "power

* Visualization of the action was attempted in Section 4–5.

† Saturated mercury vapor has a temperature of 706° at 20 psi., whereas saturated steam at 705° develops a pressure of 3206 psi.

plant" or "power station." It is instructive to visualize how energy is handled in such a plant. In one like that shown in Figure 12–1 the fuel, which would usually be coal, is first transported from storage to the combustion equipment, located in or attached to a furnace. It is burned with air, producing heat energy. This heat is mostly taken up by the water substance in the boiler as it undergoes the change from liquid to vapor. The heat is next transported to the prime mover, a turbine in this case, by convection as the steam flows through the connecting pipe. There most of the "available" heat

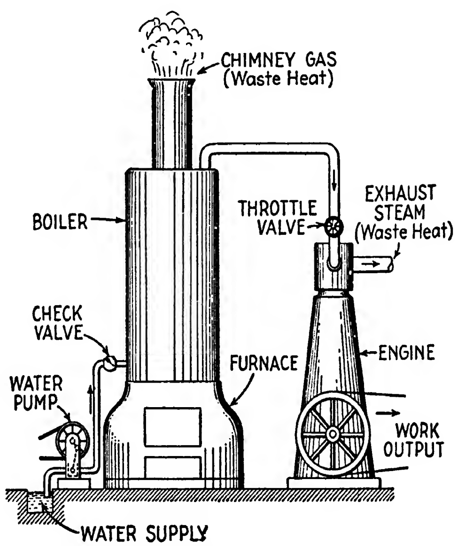


Fig. 12–2. Simplest steam power plant.

is released by adiabatic expansion of the steam, appearing next as mechanical work available as a rotating torque in the turbine shaft. By coupling the turbine shaft to a generator, fan, pump, or propeller shaft (or any other suitable power-driven rotating machine), the work output of the system can be utilized as desired.

Everywhere there is transfer or transformation of energy, and all these actions will involve some attrition of the energy. Consequently the energy finally available at the driven machine will be but a modest portion of the potential energy originally in the fuel.

The illustration depicts only part of the equipment usually found in a steam power plant, for it omits pumps, heaters, fans, controls, and numerous other items which will be found in most steam plants for the purpose of (1) completing a closed cycle for the flow of the working medium instead of wasting the exhaust steam to the atmosphere or (2) increasing the efficiency of the heat → work conversion by conserving heat that might otherwise be wasted. The simplest of all steam plants would have (1) a boiler with furnace,

(2) an engine, and (3) a pump for maintaining water in the boiler. It would be called a non-condensing steam plant since no condenser is provided. This is the proper designation regardless of the fact that the exhaust steam *is* condensed upon its contact with the cool atmosphere. A suitable supply of "feed water" is required at the water pump suction. This water is pumped into the boiler to replace that evaporated into steam. The pump is essential because unless the steam produced by the boiler is above atmospheric pressure, no work-producing expansion would be possible in the engine. The pump must increase the feed water pressure until it will flow past the non-return valve into the water space of the boiler.

This steam plant makes a virtue of simplicity; however, it wastes so much valuable heat energy that such designs can not often be defended on economic grounds. The expense of some extra equipment of a heat-saving nature will almost always be justifiable by saving in cost of fuel. How much of this extra equipment will prove to be economic depends on these factors:

1. *The extent to which the plant is actually used.* Stand-by or emergency plants must be kept simple since expensive efficient equipment would have scant opportunity to justify itself in a plant mainly standing idle. A central steam-electric station, generating electricity for public use, represents the other extreme where continuous operation is the rule. In these plants are found great quantities of auxiliary equipment whose presence assists the whole ensemble to wring more work energy out of the vapor cycle.
2. *The cost of the fuel.* The more expensive the fuel is per unit heating value, the greater will be the financial advantage of saving some of it by expenditures for heat-saving equipment.
3. *The availability of suitable feed water.* If water is scarce, then equipment to close the vapor cycle will be wanted, even if fuel is cheap, or if the plant is infrequently operated. A condenser, supplied with some available impure water (say sea water) could receive the exhaust steam and liquefy it by transferring its remaining latent heat of evaporation to the condensing water through a separating partition.* The condensed steam (called *condensate*) can then be recovered and pumped back to the boiler as "feed."

The steam condenser as a vapor plant auxiliary can be justified on both water-conserving and thermal efficiency bases. If the steam space in it is made air-tight a vacuum can be maintained. This promotes more expansion of the steam while in the prime mover, yielding better thermal efficiency for the plant as a whole. This advantage is so pronounced in turbine plants that the condensing steam turbine is customary unless (1) adequate fuel is avail-

* Usual form is tubular with condensing water inside, steam outside.

able as a waste product without other commercial value, (2) it is inconvenient or impractical to have the condenser because of its bulk, lack of condensing water, * etc., or (3) the exhaust steam may be used industrially, for heating buildings, etc.

12–4. The Rankine Vapor Cycle. Rankine's † modification of the Carnot cycle is the basis of the modern steam plant cycle, even though the Rankine

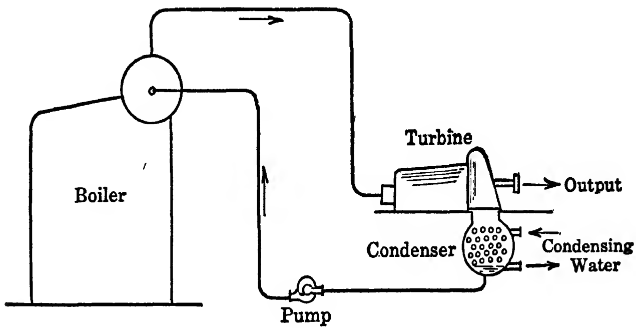


FIG. 12–3. Flow diagram of the simple Rankine vapor cycle plant.

cycle itself has been improved with the passing of time. The cycle is a sequence of processes which, if carried out on a vaporizable working medium, will transform some of the heat supplied to the cycle into an output of mechanical work. The minimum equipment required to produce the cycle is shown in Figure 12–3. It is seen to consist of (1) a boiler which receives feed water from (2) a pump, (3) a prime mover for the adiabatic expansion of the steam, and (4) a condenser to receive the exhaust steam and reduce it to the liquid state. A P–V diagram of the Rankine cycle shows that constant pressure processes are conducted in the boiler and condenser regions, while the pump and prime mover raise and lower the pressure, respectively. The efficiency of the Rankine cycle is the ratio of prime mover output to heat absorbed by the water from sources external to the cycle, i.e., from combustion of a fuel. The efficiency could be written as an equation, thus:

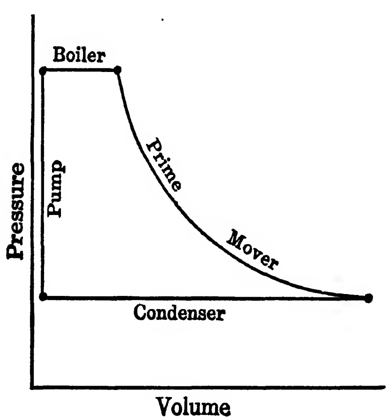


FIG. 12–4. Rankine vapor cycle.

$$\eta_R = \frac{\text{Work output of prime mover}}{\text{Heat supplied (via the boiler)}}.$$

* An example is the steam turbine locomotive.

† Professor, University of Glasgow, Scotland. B. 1820, d. 1872.

η_R will symbolize *Rankine cycle thermal efficiency*, and, of course, both numerator and denominator must be in the same physical units, say in B.t.u.

Allowing that the prime mover is an ideal steam engine or turbine, operating expansively, the work done equals the enthalpy change of the steam in

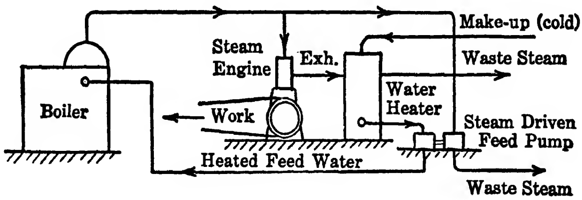


FIG. 12-5. Flow diagram of an open cycle steam engine plant.

passing through it. This is $h_1 - h_2$ B.t.u. per lb. steam. Since the condensed steam at enthalpy h_{f_2} is returned, without loss, to the boiler as feed water, the boiler is called on to supply $h_1 - h_{f_2}$ B.t.u. to each pound of steam passing through it. So the Rankine cycle efficiency becomes

$$\eta_R = \frac{h_1 - h_2}{h_1 - h_{f_2}},$$

in which h_1 = Enthalpy of steam passing from boiler to prime mover.

h_2 = Enthalpy of steam leaving prime mover after an isentropic expansion from h_1.

h_{f_2} = Enthalpy of condensate at condenser saturation temperature.

This ideal η applies to a case of complete expansion from boiler to condenser pressure. Many cases are found where this requirement is not met, but, in any event, a return to the output/input idea of thermal efficiency will generally secure a solution. For example, the steam plant shown in Figure 12-5 contains elements of the Rankine cycle, but note that a study of its efficiency must include attention to the following:

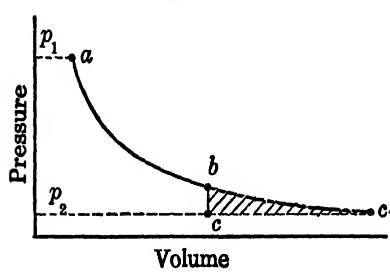


FIG. 12-6. Complete vs. incomplete expansion.

1. There is no condenser, but some of the exhaust steam is used to heat cold water for boiler feed. Some of the exhaust heat is thus conserved.
2. In an actual engine the exhaust steam will possess higher enthalpy, and the work produced will be less than for an ideal case. This may come about as the result of friction, radiation, and other heat losses—including *incomplete expansion,* which is termination of the working expansion process prior to attainment of exhaust pressure. Expansion *abc′* in Figure 12-6 is a complete, *abc* an

incomplete, expansion, both from p_1 to p_2. The incomplete expansion sacrifices the work area $bc'c$ to gain a reduction of piston displacement (tantamount to a corresponding reduction of engine size) of cc'. This practice is considered desirable in engine construction, but complete expansion is practical in the turbine field.

3. In this plant (Figure 12–5) the steam-operated pump uses some of the heat supplied to the steam, whereas in the ideal cycle it was assumed that some outside source would power the pump.

The efficiency of a steam power plant like that of Figure 12–5 is necessarily very poor, yet for small power rating it has the advantage of minimum investment and simplicity of operation.

Example 1: An ideal Rankine vapor cycle is operated between pressure limits of 150 psi. and 3 psi., both absolute. The condition of the high-pressure steam is "dry and saturated." It is desired to calculate the ideal efficiency.

After an ideal adiabatic expansion from 150 psi. to 3 psi., $s_3 = s_{150}$. Now s_{150} can be obtained by consulting the saturated steam tables for s_g at 150 psi. It is 1.5698. At 3 psi. s_g is more than 1.5698, which is to be interpreted as meaning that the steam is wet after the expansion and its entropy is of the form $s_f + xs_{fg}$. At 3 psi. s_f is .2008, and s_{fg} is 1.6855. Equating s before and after expansion,

$$1.5698 = .2008 + 1.6855x.$$

$$x = .812 \text{ quality.}$$

With this quality we next compute h_2, using h_f and h_{fg} from the tables at 3 psi.

$$h_2 = 109.4 + .812 \times 1013.2 = 932 \text{ B.t.u. per lb.}$$

Since h_1 is 1194.4, and h_{f2} is 109.4,

$$\eta_R = \frac{1194.4 - 932}{1194.4 - 109.4} = 24.2\%.$$

Example 2: Estimate the potential work lost by cutting off an expansion before completion. Assume that p_b (Figure 12–6) is 30 psi. and p_c 3 psi., and that the steam is dry and saturated at 150 psi. at point a. The quality at b is calculated in a manner similar to Example 1, using steam table data from the 150 psi. and 30 psi. entries.

$$s = 1.5698 = .3680 + 1.3312x.$$

$$x_b = .903.$$

Then

$$h_b = 218.8 + .903 \times 945.2 = 1072 \text{ B.t.u. per lb.}$$

The volume of the steam at b is $x_b v_g$ nearly, or $.903 \times 13.8 = 12.5$ cu. ft. per lb. By dropping the pressure from 30 to 3 psi. at constant volume (line bc) the engine is enabled to produce not only $h_a - h_b$, but, in addition, $12.5(30 - 3) \times 144$ ft. lbs. of work. This is because the exhaust can occur at 3 instead of 30 psi. But note that the work still is less than if a complete expansion to c' had been carried out. Work

done per pound of steam, following *abc*, is (1194.4 − 1072)778 + 12.5(30 − 3)144 = 143,500 ft. lb. per lb. of steam, whereas that for expansion *abc'* is

$$(1194.4 - 932\ ^*)778 = 204{,}200 \text{ ft. lbs. per lb. of steam.}$$

Evidently 204,200 − 143,500 or 60,700 ft. lbs. is the loss of work per lb. of steam caused by incomplete expansion. The volume of the 3 psi., 81.2% dry steam at c' being 95.3 cu. ft. per lb., there is a reduction to 12.5/95.3 or roughly 13% of the cylinder size necessary for complete expansion. That it would be desirable in most cases to save 87% of engine bulk for a price of 30% of the available work (probably representing about 4% of the heat input to the cycle) seems almost obvious.

To continue the output/input analysis, $h_c = h_a - W/J$, therefore enthalpy h_c is 1194.4 − (143,500/778) = 1010 B.t.u. per lb.

$$\eta_R(\text{for the incomplete expansion cycle}) = \frac{1194.4 - 1010}{1194.4 - 109.4} = 17.0\%.$$

Summarizing the above: the engine of the cycle can be made approximately $\frac{1}{8}$ as large for a drop of $\frac{1}{3}$ in efficiency by employing an incomplete expansion cycle.

12–5. Other Cycles. When means were sought to increase the Rankine vapor cycle performance, it was natural to employ even higher h_1's and lower h_2's (of the η_R equation). High initial enthalpy is secured by using higher boiler pressures, degree of superheat, or both. Likewise for minimum h_2 high vacuum condensers are indicated. The initial condition finally reaches a peak of development limited by metallurgical considerations—high temperature alloys for the boiler, superheater, piping, etc. Existing condensing water temperature (river water, lake water) does likewise for the minimum final state.

New variations of the vapor cycle then appear, particularly in the steam-electric station where high degree of utilization of equipment justifies many a pioneering move. Most of the large central power stations of the present time operate on a cycle known as the *regenerative cycle*, so called because of the method of regenerating the thermal potential of the working medium (feed water) in order to condition it for the boiler. Nearly fifty years ago Cotterill saw that the extraction of some of the steam from an engine, for the purpose of bringing the boiler feed water nearer to the saturation temperature of the boiler, would result in considerable thermodynamic gain over the simple Rankine cycle. This idea was first applied to reciprocating steam engine plants, but its present widespread use did not come until the advent of the high-capacity steam central station. The reason for this is to be sought in the expense of the regenerative equipment, which could not be justified in small plants.

The principle of this cycle is shown in Figure 12–7. As the steam passes through the turbine, a small quantity is extracted through an opening pro-

* From Example 1.

vided between stages in the casing. Of course this steam is somewhat lower in thermal potential than the throttle steam, but is definitely superior to the exhaust state. Its exact condition is contingent upon the location of the extraction opening. This steam is conveyed through a pipe to a heat exchanger in which it condenses, giving up its heat to the feed water which is travelling from the condenser back toward the boiler. In this way the h_{f_2} of the η_R equation may be increased and thermal efficiency thereby benefited. The reason for the efficiency gain of the regenerative over the Rankine cycle lies in the fact that the steam, as extracted from the turbine, has performed a certain amount of work, but whereas the percentage of the available heat converted

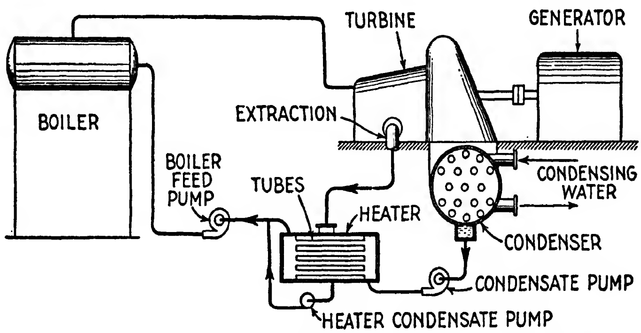


Fig. 12–7. Principle of regenerative feed water heating.

into work before extraction is considerable, the percentage decrease in the total heat of the steam is small, and hence, by extracting it before it reaches a low thermal potential, it has remaining in it a high degree of availability as a feed water heating medium. This factor becomes of more and more importance as boiler pressure increases.

In large electric stations, several points of extraction are used and the feed water is "regenerated" thermally almost to the boiler saturation temperature. The cost and complexity of producing the vapor cycle is greatly augmented, but the performance may be 25% better than for the plain Rankine cycle operating between the same terminal conditions.*

Other modifications of the vapor cycle introduce (1) resuperheating the steam, (2) use of two vapors, mercury and steam, and (3) various special combinations possible in the industrial field where both power and process steam are wanted.

Resuperheating is practiced if the expansion attempted in the turbine is so much that the steam becomes unduly burdened with water of condensation. Too much of the latter has a strong tendency to erosion of internal portions of the turbine. Rather than sacrifice the possible advantages of high expan-

* If η_R is 20%, and the regenerative cycle has 25%, then it is $(25 - 20)/20$ or 25% better.

sion ratio, sometimes all the flow of steam is extracted from the turbine after partial expansion, resuperheated, then returned to the turbine to continue the expansion. Such an arrangement has been termed the *"reheating cycle."* It has been but seldom employed, but when found the installations are representative of advanced heat power engineering, i.e., large high-pressure,

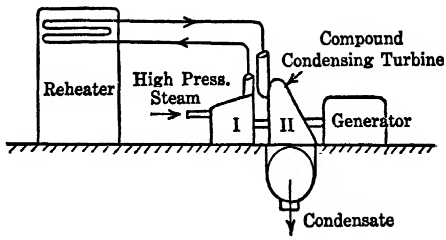


FIG. 12-8. Modification for reheating the steam. Resuperheating the steam intermediately prevents its acquiring excessive moisture in the low-pressure turbine (II).

efficient steam-electric stations. The principle may be combined with feed water regeneration to make a "reheating-regenerative" cycle.

Although water vapor has been the standard working medium of the commercial power plant employing an external combustion cycle, it was recognized long ago that water vapor had physical properties not altogether desirable at either the high temperature or low temperature end of the cycle, but at no time has any vapor been discovered which would be more satisfactory at both extremes of the expansion range than steam. The use of two vapors in series makes it possible to keep an extensive temperature range and

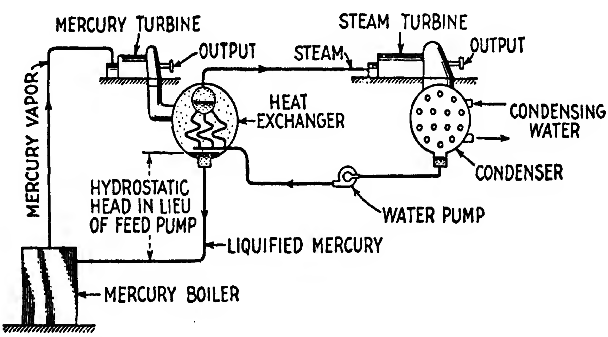


FIG. 12-9. Principle of the binary-vapor cycle.

eliminate several of the disadvantages attending the use of a single vapor. The mercury-steam is the only *binary vapor cycle* operated on a commercial scale at present.

When some low-pressure steam is needed in addition to mechanical power, a non-condensing engine or turbine, or an extraction turbine, is frequently employed in order to meet both needs from the same machine. The ingenious

designer may find varied opportunities in selecting equipment and flow plans in which a nice balance is established between needed power and process steam. There is no standard *"industrial cycle"* as such, but each installation is some interesting modification of the basic Rankine vapor cycle.

12–6. Steam Locomotive. One form in which the external combustion cycle is frequently built is the common steam locomotive. Figure 12–10 is a diagram illustrating the locomotive as a complete power plant. It is a non-condensing, variable speed, steam engine plant. The power output is represented in the locomotive's ability to exert a drawbar pull on a moving train.

The main function of a locomotive is to exert a pull at the coupler. This it accomplishes by applying a rotative effort to driving wheels which rest on the track. Since the drawbar pull cannot possibly exceed the weight on the drivers multiplied by coefficient of friction between driver and track, it is seen that there is not the same urge for light-weight construction in a locomotive as in the power plant, for example, of the transport airplane. The elements of the locomotive consist, in part, of a frame and running gear, the latter consisting, in part, of a number of wheels called drivers, to which the propulsion is transmitted in the form of a rotation. The reactions developed at the wheel bearings are transmitted through the frame to the coupler. Another element is the source of power. In the common steam locomotive this is an expansion *steam engine* receiving steam from a boiler, and exhausting against atmospheric pressure. By means of a throttle valve located on the steam line leading to the engine, and by valve gear adjustments, the power developed in the engine cylinder is varied to suit the needs. The cylinders are mounted with axes longitudinal, and in line with the drivers. Steam locomotives are two-, three-, or four-cylindered, the most common arrangement being two cylinders. Where three cylinders are used, the third cylinder is mounted under the boiler, and between the other two. Although compound-expansion locomotives have been built, the common locomotive is a single-expansion type. The connecting rod which extends from crosshead to one of the drivers is enabled to transmit propulsion equally to all drivers by use of the side rod, to which the crankpins of all drivers are connected.

Typical dimensions of a freight locomotive are: boiler pressure, 245 lbs. per sq. in.; cylinders, 25-in. bore by 34-in. stroke; total weight 400,000 lbs.; weight on drivers, 260,000 lbs.; diameter of drivers, 69 in.; drawbar pull, 64,000 lbs.

Another essential element is the *boiler* in which the steam is raised. Through many years of evolution, the boiler has become standardized on a horizontal fire tube type, with completely water-cooled furnace. The furnace end is placed at the rear of the locomotive, and the products of combustion pass forward through the tubes to the smoke box, from whence they are discharged upward to the atmosphere through a short stack. This type of

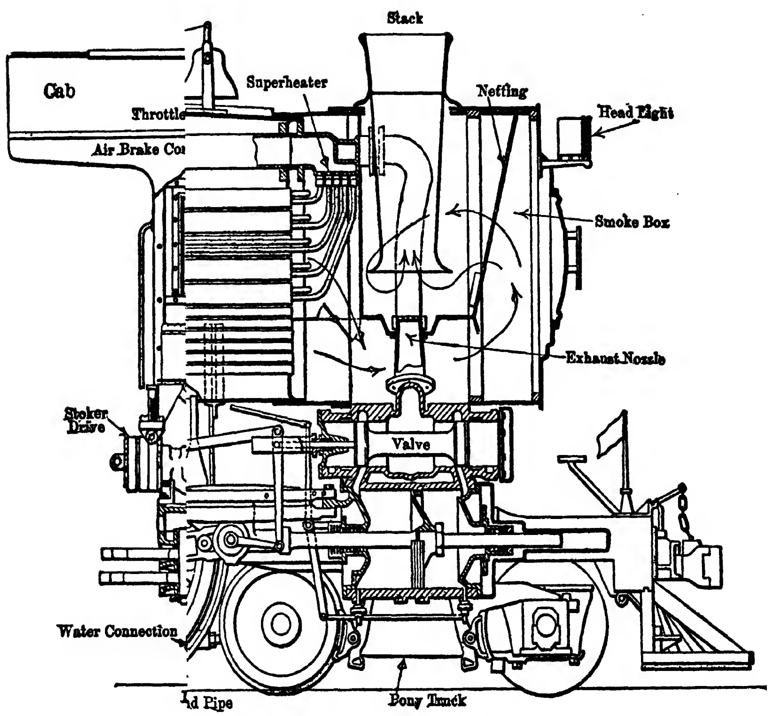


FIG. 12–10. Locmwood & Hirshfeld, published by John Wiley & Sons, Inc.)

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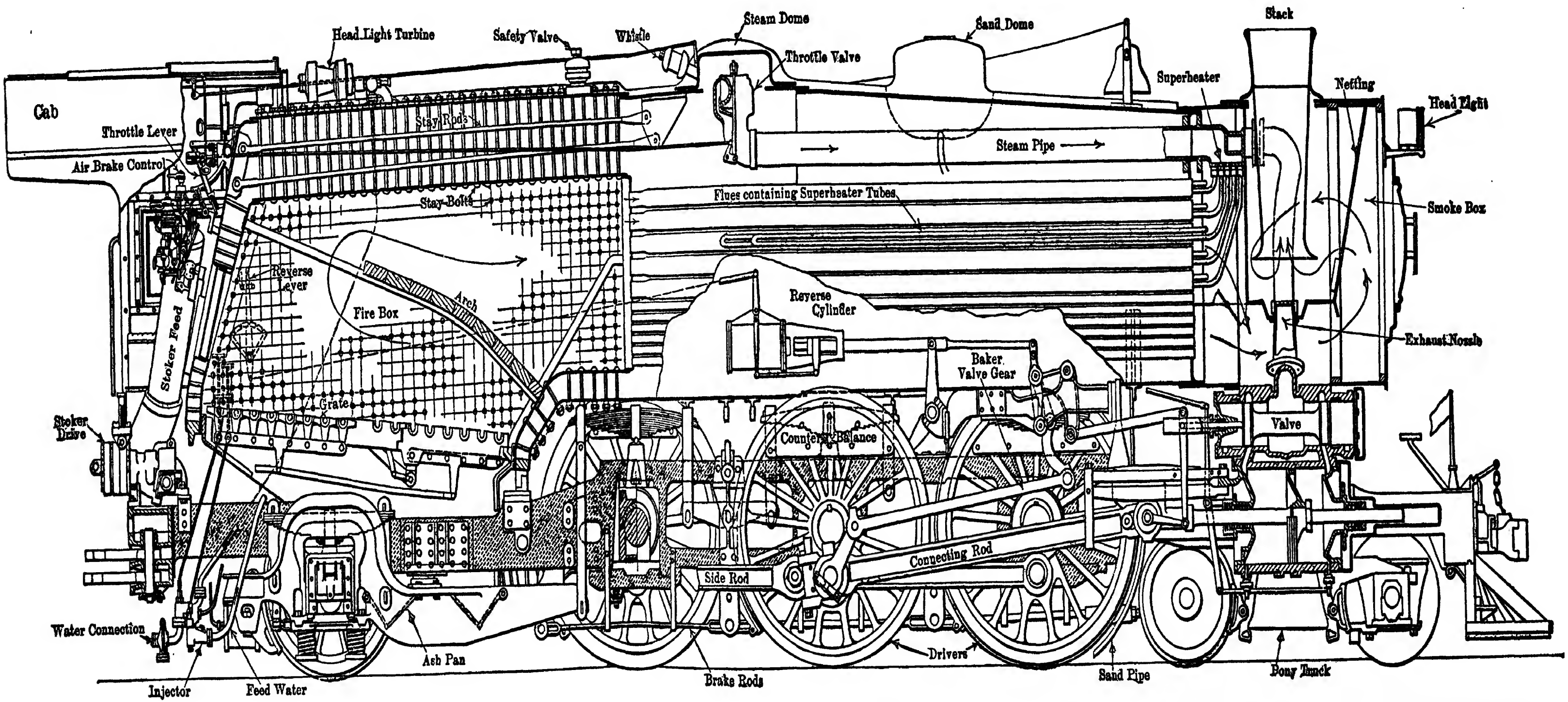

Fig. 12-10. Locomotive power plant—Pacific type of passenger locomotive. (Reprinted by permission from *Heat Power Engineering* by Barnard, Ellenwood & Hirshfeld, published by John Wiley & Sons, Inc.)

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boiler, in large sizes, has tubes of sufficient size so that superheater elements may be installed in them, for the use of superheated steam has greatly improved the performance of the steam locomotive. Due to the use of stay-bolt construction to hold the inner and outer shells of the water leg around the furnace, the boiler pressures carried are limited to approximately 250 lbs. per sq. in. The combustion systems include oil firing, hand firing of lump coal on grates, and stoker firing of coal. Draft is obtained by aspirator action of the exhaust discharging into the stack. As the locomotive is non-condensing in operation, sufficient supply of water for a run must be carried in the tender tank. From the tender tank it is pumped and heated by an *injector*, which is the most common means used to supply feed water to a locomotive boiler.

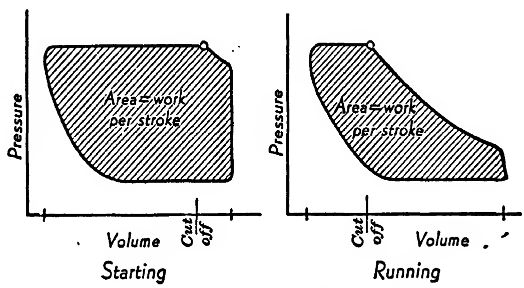


FIG. 12–11. Cylinder pressure-volume diagrams for two load conditions.

The horsepower capacity of the steam locomotive varies from 1200 to 3000, and occasionally is somewhat higher. The boiler efficiency averages from 40% to 50% under load. While this may seem rather low compared to the performance of stationary steam generating boilers, the conditions of operation of locomotive boilers require a compromise between efficiency and practicability for motive service. The overall thermal efficiency of the steam locomotive is about 4%. For starting purposes, the valve gear of the steam locomotive is set so that the steam is cut off from the cylinders relatively late in the stroke. This gives considerable more work available per revolution than when the cut-off is advanced to a more normal and economical point. This is illustrated by the pressure-volume diagrams showing conditions in the cylinders at starting and running. The mean effective pressure may be made nearly equal to full boiler pressure with very late cut-off; however, this operation is so wasteful of steam that it is employed only for starting heavy loads.

The complete locomotive includes, in addition to the elements mentioned above, an enclosed cab containing the principal train and locomotive controls conveniently arranged for operation by the engineman. The locomotive must have air-brake equipment and control, as well as lighting and various safety devices.

The steam locomotive has been evolved and perfected through approximately a century of mechanical progress. The design has become standardized, with the result that in spite of size, weight, and power, the steam locomotive is relatively cheap at present.

12-7. Central Power Station. A steam central power station is a vapor cycle plant in which combustion of a fuel provides the heat from which the

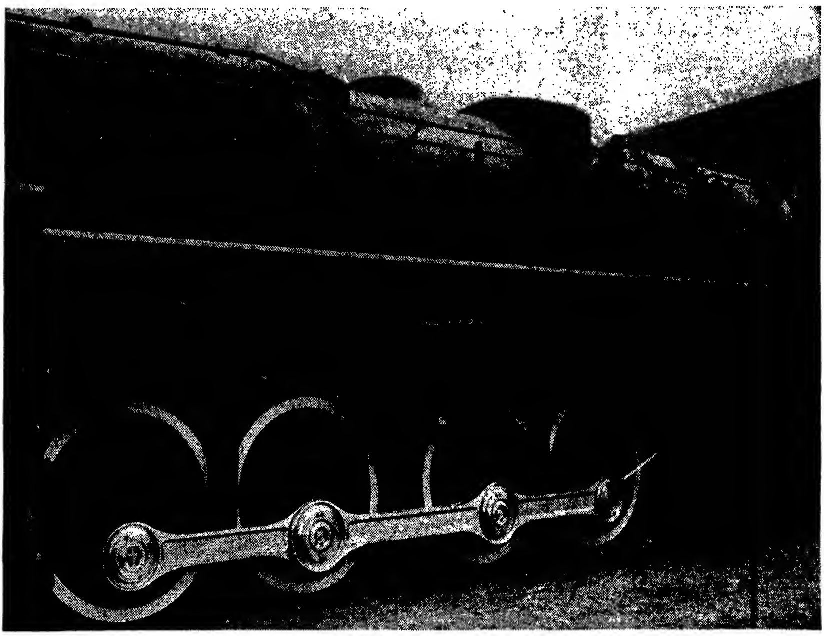


FIG. 12-12. Steam locomotive. Geared steam turbine drive. (Courtesy Baldwin Locomotive Works.)

cycle derives mechanical work. This, when converted to the electrical form, is readily sent, via transmission lines, to the private systems of the numerous customers who purchase the electrical energy (usually metered and paid for as kilowatt hours). Continuous use of the equipment and other operating features of this field of applied energy justify much accessory equipment whose use promotes high thermal efficiency. The equipment when assembled in a permanent way, suitably housed, and connected together with pipes, wires, ducts, etc., as may be needed, is called a "station." It must be near a large natural water supply (for condensing) and be available to a commercial delivery of fuel, usually a railroad. From it transmission lines extend to the vicinity where the customers' systems are located. Then by multiple branching circuits (the secondary distribution system) a connection is made with each customer's system so that all are in electrical contact with the central station. The central station is illustrative of the fact that large-scale produc-

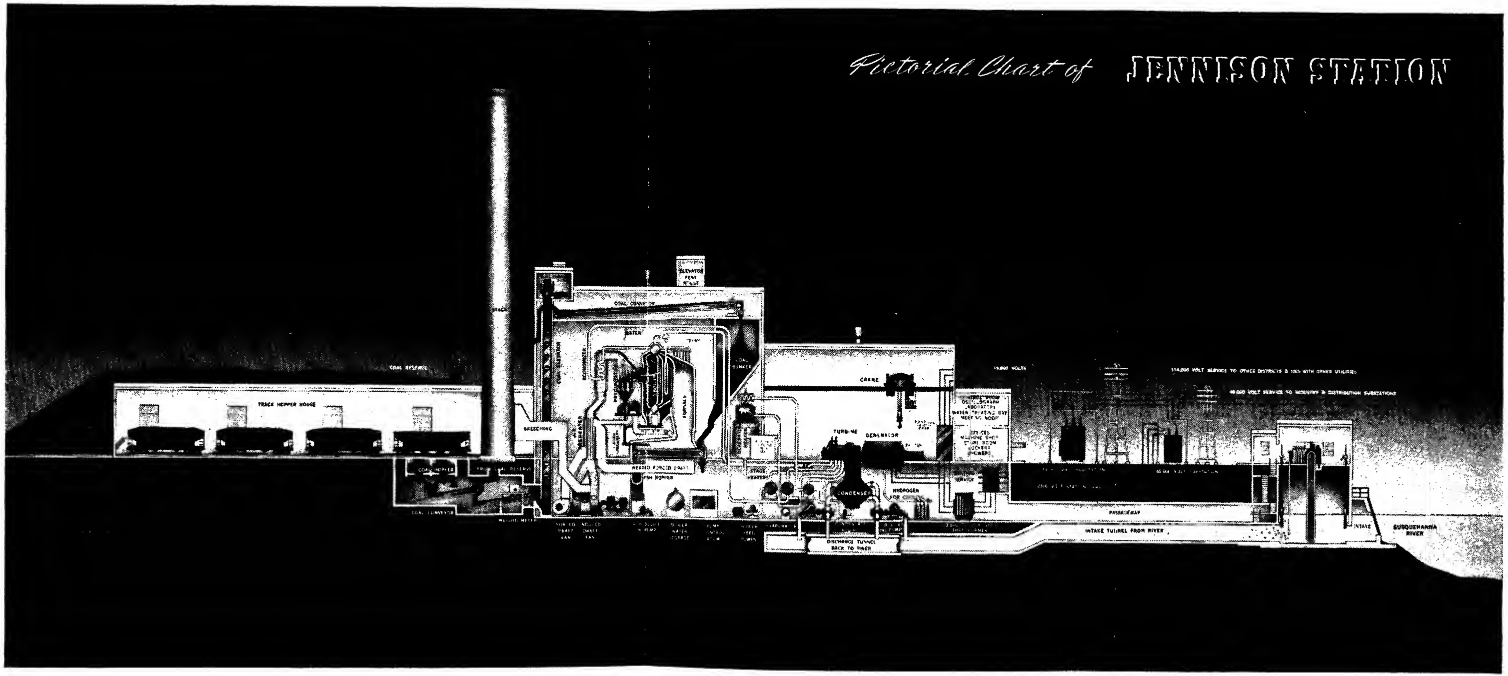


Fig. 12–13. Diagram of a modern coal-burning central power station. Jennison Station of the New York State Electric and Gas Corp. (Courtesy New York State Electric and Gas Corp.)

To face page 318.

tion is the most economical way of providing electrical energy to the individual consumer, for energy thus produced may usually be sold to the individual for less than he could generate it himself, in spite of the fact that there will be transmission losses on the line between the central station and the customer.

A typical steam central station will consist primarily of a boiler, turbine, and condenser—the boiler to produce steam, the turbine to utilize the steam for the production of mechanical power, and the condenser to receive the exhaust steam and reduce it to liquid for returning to the boiler, and at the same time provide a vacuum for the more economical operation of the turbine. A generator directly connected to the turbine effects conversion of mechanical into electrical energy with a minimum of loss. The boiler-turbine-condenser group must be serviced by auxiliary equipment which might, for clarification, be grouped under two heads—that auxiliary equipment which might be said to be connected with the flow of air or flue gas in the combustion portion of the plant, and that having to do with the proper conditioning of the condensate produced by the condenser, so that it possesses the requisite heat, pressure, and purity for boiler feed. Included in the first of these groups will be such equipment as draft fans, stokers, air preheaters, etc., while in the second group there would be found feed-water heaters and pumps, feed-water treatment, etc. Between the generator or generators and the outgoing transmission lines, there is contained in the central station no inconsiderable amount of electrical equipment needed for the control and electrical protection of the generators, the supply of auxiliary power, and switching.

Figure 12–13 is seen to be typical of the type of station just described. This illustration is, of course, schematic, but conveys a good idea of the equipment, its relative location, and the flow of the working medium. The coal, which is received in railway cars, is carried to a storage bunker from which it flows by gravity to a travelling grate stoker. Products of combustion flow from the furnace through the boiler, cinder trap, economizer, air preheater, fan, breeching, and thence into the stack which discharges them to the atmosphere. After receiving a superheat, the steam that is generated (650 psi., 825° F) flows to the turbine where it is expanded to a pressure of 1.5 in. Hg abs. Entering the condenser at that pressure, the exhaust is condensed and starts on its return journey to the boiler. After passing through several regenerative heaters, and receiving deaeration and make-up of leakage losses, if any, this condensate becomes boiler feed water. It has received from the boiler feed pump a pressure sufficiently high to enable it to enter the high-pressure region. Each half of the condensing surface has its own circulating pump which draws water from an intake tunnel. Cross-over headers are installed so that one pump can, if needed, serve the whole condenser. The 30,000-kw. alternator generates 13,800-volt energy, which is transformed to 114,000 volts for cross-country transmission. One can find here the elements

previously shown in Figure 12–7. The purpose of many of the other items of equipment will appear in Chapter 15.

12–8. Marine Power Plant. A steam power plant, with features quite different from those just described, is the propulsion plant of a steamship. The distinctive requirements for this service are these. The ouput is mechanical work in the form of a torque applied to a rotating shaft on which the water propeller is mounted. These propellers function best at low speeds so the torque is generally quite large and the equipment massive. Reciprocating steam engines are direct-connected to the propeller shaft but steam turbines with their high-speed characteristics are usually joined to the propeller shaft by speed-reducing gears. A closed cycle is always used because river and sea water are unfit for continuous boiler feed. Of course, there can be no interconnection and dependence on any other power plant, so the ship's power plant must be completely self-sufficient. Auxiliaries are more often steam-driven than in stationary plant practice. The highly advanced steam cycles mentioned in Section 12–5 are not seen in this field. Space is limited, especially horizontally, therefore whenever a piece of equipment can be built in either horizontal or vertical pattern, the latter is likely to be selected for shipboard service. Although simplicity and compactness are sought, high thermal efficiency is also wanted since the less the fuel required to be burned on the voyage between ports the more the cargo capacity.

Steam power was first applied to the propulsion of ships in the United States in the 1810–1820 decade. A hundred years later the fire-tube boiler and the triple expansion steam engine reached their peak of development in this field, then gave way rapidly before the superior features of the steam turbine and the water-tube boiler. Turbine drive was almost standard for the great ship building program of World War II wherever steam was used. Some engine plants were installed simply because sufficient turbines and reduction gears could not be manufactured to meet the needs of the moment.

The marine plant shown in Figure 12–14 is diagrammatic only. The figure shows a plan of the main propulsion unit. Auxiliaries such as blowers, burners, evaporators, etc., are omitted, as are also the many pieces of ship's service equipment such as air compressors, fire pumps, etc., most of which are usually located in the region shown.

The boilers are somewhat like that of Figure 13–8. Fuel oil is especially suitable for marine plants as the firing equipment is simple and the fuel is readily pumped from storage tanks to the burners. Many vessels are so equipped, especially those of the Navy, but coal is also in common usage in the maritime world. Saturated steam from the boiler drum is led through superheaters and carried by the main steam header. Two boilers are shown in Figure 12–14 but ships will be found having many more than this number. Older ships with fire-tube boilers sometimes had from sixteen to twenty-four

boilers, but newer designs with water-tube boilers have fewer because of the greater capacities in which that type can be built. The steam header is connected to a throttle stand where the hand-operated throttles for forward and reverse operation are located. Normally steam passes to the high-pressure turbine where it is partly expanded, then flows through a cross-over pipe to the low-pressure turbine. The steam outlet of this turbine is connected to the steam condenser so a very low pressure exists at the exhaust of this section of the turbine. A backing turbine of low power is mounted on the same shaft.

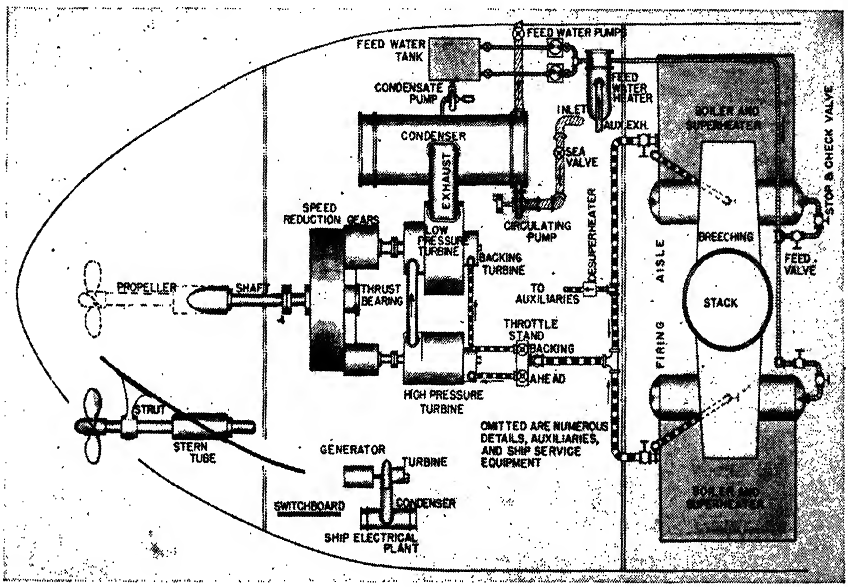


FIG. 12–14. Marine plant layout.

The nozzles and blades are arranged so that the shaft will be driven in reverse direction when steam is admitted to this turbine. Its exhaust takes the same path to the condenser as would that of the normal forward unit.

The main turbine shafts are connected to a double reduction gear drive. The propeller speeds being in the lower hundreds of revolutions per minute, and turbine speeds in the lower thousands, this speed reduction must be of the order of 10:1 or more. Reaction of the water against the propeller blades sets up a powerful forward thrust that is transmitted via the propeller shaft to the hull of the ship. Since the shaft is in motion a *thrust bearing* must be provided to transfer the thrust. The enormous force involved requires a special bearing, one of the most frequently used being the Kingsbury bearing. This is a babbitted type of bearing having very low friction although carrying a high pressure from the rotating to the fixed face. This is made possible by dividing the bearing surface into a number of pivoted segments which tilt in

such a way as to scrape a wedge of oil between them and the rotating collar which is mounted on the propeller shaft. The shoe which carries the segments is stationary and securely attached to the framework of the ship. Shoes are needed on either side of the collar, one for thrust ahead, the other astern.

The exhaust steam passes directly from the low-pressure turbine into the condenser. Inside the condenser are many small tubes through which sea water is pumped. These quickly soak up the heat in the steam. Although large volumes of exhaust steam continuously pour into the condenser, condensation is quick and complete as long as plenty of sea water is kept in circulation through the tubes. A small stream of condensate is pumped out of the condenser into a feed tank. Boiler feed-water pumps, usually steam-driven, take this water and raise its pressure higher than the boiler's. When a feed valve on the boiler supply line is opened the feed water will be able to flow in under the pressure it acquired at the pump. On its way to the boiler, the feed is heated by exhaust steam collected from the various steam-driven auxiliaries such as feed pumps, blower turbines, etc.

It is obvious that the problems of steam locomotive design are altogether different from those of the central station. Although not equally obvious, it is nevertheless true that the problems of marine propulsion design are quite different from those encountered in the central station or any other stationary steam plant. It is a highly specialized field of applied energy.

PROBLEMS

1. Diagram the equipment of a simple steam plant having a vertical fire-tube boiler supplying steam to an engine. A motor-driven pump is to pump feed water from a tank (supplied from a city water line) into the boiler.

2. Diagram the simple steam plant often sold as a children's toy. An engine is usually mounted on a boiler having one vertical flue inside the boiler shell. Heat is supplied by an alcohol lamp and there is no continuous feed system.

3. Sketch (side view) a locomotive with the object in mind being a functional diagram of the locomotive as a steam power plant.

4. Draw, to scale, a Rankine vapor cycle (similar to Figure 12–4) for a boiler generating dry and saturated steam at 50 psi. abs. Condenser operates at a saturated steam temperature of 150° F. Expansion is complete and isentropic. The expansion line may be approximated after the initial and final points are located. Diagram is to represent one pound of H_2O, scales 1 in. = 10 cu. ft., 1 in. = 10 psi.

5. Draw the Rankine vapor cycle (Figure 12–4) to scale for the following conditions. One pound H_2O. Boiler pressure 150 psi., 50° superheat. Condenser 25 in. Hg vacuum. Scale 1 in. = 20 cu. ft., 1 in. = 20 psi. Equate entropy of the superheated steam to that of exhaust steam to find quality and volume of steam at the end of expansion.

6. Calculate the ideal thermal efficiency of the cycle specified in Problem 4.

7. Calculate the ideal thermal efficiency of the cycle specified in Problem 5.

8. What is the ideal thermal efficiency of a complete expansion steam turbine plant receiving steam at 80 psi. dry and saturated, exhausting at atmospheric pressure?

9. Calculate η_R for steam plant terminal conditions as follows: Boiler 400 psi. abs. 500° F. Condenser 27 in. vacuum. Barometer 30 in.

10. A steam engine plant has a boiler which produces dry and saturated steam at 75 psi. gage. The steam is isentropically expanded by the engine until pressure drops to 15 psi. gage. Then the engine exhaust valve opens and pressure sinks to atmospheric pressure (15 psi.) at constant volume (see Figure 12–6).

a. Calculate the quality at c and c'. Find ratio V_b/V_c'.
b. Find the foot-pounds of work done by the engine per pound of steam.
c. By what percentage does this use of incomplete expansion reduce the ideal available work?
d. Sketch the Rankine cycle. No scale, but label completely with the data given or calculated.

11. Solve a problem equivalent to No. 10, except that boiler conditions are 130 psi. abs., 98% quality, and exhaust takes place when the pressure reaches 20 psi. abs.

12. Suppose, in Problem 10, exhaust begins when $V_b/V_a = 4.0$. What release pressure, p_b, is indicated? Note: By trial, using the Mollier diagram, find on the s_a line that point where the quality yields the desired V_b. Then read the pressure at that point.

13. Suppose, in Problem 11, exhaust is wanted when the expansion ratio (i.e., V_b/V_a) is 5.0. What release pressure, p_b, is indicated? (See note, Problem 12.)

14. Construct a flow diagram showing the connections in an industrial power plant described as follows. A boiler is connected to a "steam header." From this header high-pressure steam is taken to the factory; also, with another connection to the header, high-pressure steam is brought to a steam turbo-generator—non-condensing type. The exhaust line from the turbine also leads to the factory to furnish needed low-pressure steam. Condensation returns from the factory into a feed-water tank in the boiler room, from which a motor-driven boiler feed pump draws its suction. The condensate return is only about 90% of the steam sent out, i.e., some of the factory operations absorb (or waste) about 10% of the steam received.

15. A steam locomotive under average load hauls a train at 40 mi. per hr., with a drawbar pull, at the tender, of 37,400 lbs., consuming, meanwhile, 190 lbs. of coal per min. Heating value of fuel 11,500 B.t.u. per lb. (a) Calculate the locomotive horsepower, (b) find the overall thermal efficiency.

16. The steam pressure of a certain two-cylinder locomotive is 150 psi. gage. The engine cylinders have a bore of 25 in., driving wheels 6 ft. in diameter. The cylinders are double-acting, and boiler steam is admitted on the average for 5 in. of each piston stroke. Assuming the steam dry and saturated as it enters the cylinders at boiler pressure, estimate the gallons of water necessary in the tender tank at the start of a 100-mile run. Allow 25% extra water for margin of safety.

17. Data on a D. L. & W. locomotive are as follows: Two cylinders 27 in. × 32 in., 77-in. driver diameter, boiler pressure 250 psi. gage, superheat (est.) 150°, tender capacity 12,000 gals. Assuming cut-off of boiler steam from the cylinder when the piston has travelled $\frac{1}{6}$ of its stroke, estimate the possible mileage between refills of the tender tank, allowing a 2000-gal. reserve.

18. Draw a flow diagram of the steam locomotive, separating the different elements as though drawing for a stationary plant.

19. A small portable steam-engine plant is set up to pump water from a flooded region. The engine is belted to a pump which works steadily, pumping 750 gals. per min. against 20 ft. head. Feed water for the boiler is flood water put into the boiler by an "injector." Boiler is handfired with coal of about 10,000 B.t.u. per lb. heating value. Diagram the pumping plant. This steam plant may be expected to have a thermal efficiency of 4%, and the mechanical-hydraulic efficiency of belt and pump is estimated at 60%. How much coal would you contract for per week's operation?

20. A marine plant (Rankine cycle type) with a single cylinder turbine provides 5500 shaft hp. Operating between 500 psi., 650° F, and 27 in. Hg vacuum, it can be expected to reach 75% of the ideal performance possible with these steam conditions. Boiler thermal efficiency 80%. Auxiliaries consume 12% of boiler output. Estimate the overall efficiency of this steam plant—fuel oil to propeller shaft.

21. Ships' propeller shafts may have torque meters. Given the measured torque at 200 rpm. as 165,000 lb. ft., mechanical efficiency of the gear box 98%. Find the shaft horsepower delivered by the ship's turbines.

22. A steam locomotive operating at 175 psi., 96% quality, releases the steam after an ideal expansion to 45 psi. What was the ratio of expansion at release? (Hint: Find the quality at release, using the Mollier Chart and isentropic expansion. With it calculate V_b.)

23. The turbines of a marine power plant expand steam from 300 psi., 500° F, to 3 in. Hg abs. What is the ideal Rankine cycle efficiency?

24. A ship is driven at cruising speed by a 220,000 lb. ft. torque on the propeller shaft. Rpm. 175. Assume that the overall efficiency of the power plant from fuel to propeller shaft is 22% based on higher heating value. How many gallons of fuel oil will be burned on a 5-day voyage? 18,000 B.t.u. per lb., 20° Baumé oil.

25. The stated efficiency of a steam central station is 28%, coal pile to transmission line. How many kilowatt hours are sent out per ton of West Virginia semi-bituminous coal burned?

CHAPTER 13

Steam Generation

13-1. Elementary Steam Generator. The generation of steam at a suitable pressure has been shown to be one of the essential processes of the vapor type external combustion cycle. What principles of applied energy are involved and what is the form of the equipment used in this process?

Suppose that steam is to be generated at some pressure, say 100 psi. gage, and superheated 100° F. The raw materials from which this steam can be "manufactured" are (1) feed water, (2) fuel, and (3) oxygen (air). The figure (13-1) illustrating steam generation from these raw materials is, of

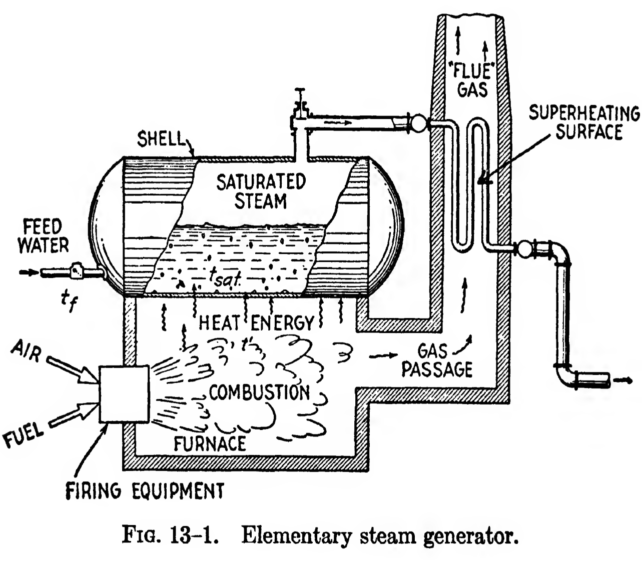


FIG. 13-1. Elementary steam generator.

course, only schematic. Actual details will follow. Primarily a steam generator is a *boiler*, shown as a simple cylindrical shell, externally heated. This boiler cân produce saturated steam only. To obtain superheat the saturated vapor must be led away from the saturation region and heated further in a *superheater*. Often a person will speak of a steam generator, which includes a superheater, as a "boiler," but this is taking some liberty with the exact meaning of the term. A large quantity of heat energy is required to generate the steam for a useful power cycle. It is produced by *combustion* and given

to water by a combination of convective, radiant, and conductive heat transfer. Bubbles of steam appear against the heated boiler shell. Growing larger, they become detached and rise through the water to break into the steam space above.

A boiler in operation represents a continuous action. Combustion is supported by a steady inflow of fuel and air to the *firing equipment* which is connected to or set into a *furnace.* Here occurs the combustion which liberates the fuel's heating value as dynamic high temperature heat energy. The enclosure of the furnace and boiler is called the *setting.* It isolates the high temperature region, also frequently supplies support for the boiler. It is not practical to transfer all the heating value of the fuel, but 60% to 90% is gotten into the water. The remainder is wasted, mostly in the hot products of combustion which are discharged to the atmosphere.

A survey of steam generation would include:

1. Steam boilers—which receive feed water continuously and deliver saturated steam. They are built in many patterns, a few of which are illustrated here.
2. Superheaters—these being surfaces necessarily external to the boiler but often enclosed in the same setting.
3. Feed water—water properly prepared for entering a boiler to take the place of that which is evaporated in the generation of steam. Although the instantaneous flow of feed water is not necessarily equal to the rate at which steam is boiled off, the total amount fed over a considerable period of time must equal the evaporation plus such loss as blowdown and steam released through the safety valve. To be suitable for the service, feed water should be at a pressure enough above that of the boiler contents, so that it flows readily into the boiler when the feed valve is opened. It should have chemical purity to the required degree, and be heated to a temperature as near that at which the boiler operates as is economically feasible.
4. Furnace—the region where combustion takes place and high temperatures are developed.
5. Firing equipment—whose function is (a) to proportion air and fuel in suitable ratios, and at the rate needed to produce heat for continuous steam generation, (b) to mix the air intimately with the fuel, and (c) to carry or inject the mixture into the region of combustion. Examples are burners for oil, gas, and pulverized coal, mechanical coal stokers, grates, etc.
6. Auxiliary and accessory equipment—which may be used to service, regulate, or protect the above. A glance through Chapter 15 will illustrate this category.

Reverting to the subject of heat transfer, note the temperatures on Figure 13–1. Furnace temperature is t'. Combustion may give the products an initial temperature of about 2000°, but this decreases as heat energy is transferred through the boiler heating surface. Final gas temperature may be as low as 500° to 900° F. At a pressure of 100 psi. gage, saturation temperature is 338° F. The feed water flows in at a lower temperature t_f, say 90° F, but it is quickly intermixed with saturated water and steam bubbles. It is usually

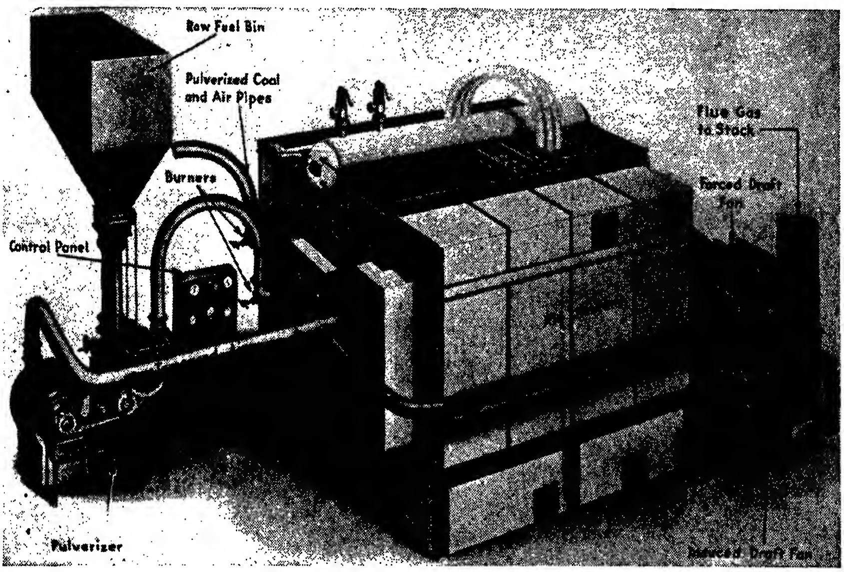


FIG. 13–2. Steam-generating unit. (Courtesy Babcock and Wilcox Co.)

considered, in heat-transfer studies, that the boiler water temperature *is* saturation temperature. Boiler heat transfer is, therefore, a case of heat release from a variable temperature t' to a constant temperature t_{sat}. Both gas and steam temperatures vary in the case of superheater surface.

Steam generation in a boiler is a constant pressure process. No mechanical work is involved, so the heat absorbed equals the gain of enthalpy. The steam's enthalpy can be taken from tables. So also can that of the feed water, via the usual assumption that it has the heat of the liquid of saturated water at t_f. In the following example we shall compare the heat energy of a pound of coal with that needed to generate steam. The usual reference standard in this field is the higher heating value.

Example: Use 100 psi. gage steam pressure, 100° superheat, 90° feed water, 12,500 B.t.u. per lb. fuel heating value. How much heat must be added, per pound of steam, (a) in the boiler, and (b) in the superheater? (c) How much steam will a pound of coal generate if 85% of the heat units are realized and transferred?

From tables one obtains the following:

$$h_g = 1189.6 \text{ B.t.u.}; \quad h(@\ 338° + 100°) = 1245.5; \quad h_{f_{90}} = 58.0 \text{ B.t.u.}$$

(a) Enthalpy gain in boiler = 1189.6 − 58.0 = 1131.6 B.t.u. per lb.
(b) Enthalpy gain in superheater = 1245.5 − 1189.6 = 55.9 B.t.u. per lb.
Total gain of enthalpy is 1187.5 B.t.u. per lb. steam.
Transferred heat per pound of coal burned = 12,500 × 85% = 10,630 B.t.u.
(c) Water evaporated and superheated per pound of coal = 10,630/1187.5 = 8.96 lb.

In retrospect we see that this ratio depends on steam conditions, feed temperature, heat losses, and fuel heating value.

Figure 13–2 is included to show where many of the elements of a *steam generator* appear when assembled as a unit. The boiler itself is well-nigh enclosed by the brick setting, as is also the superheater. Firing equipment consists of burners which blow prepared coal into the furnace. The coal is prepared in a pulverizer where it is ground so finely that it may then be floated to the burner on a little of the combustion air. The air is prepared for the burner by receiving some preheat and plenum. This yields a hotter flame and at the same time recovers some waste heat, for the air preheater is simply a means of transferring heat from flue gas to air.

13–2. Boiler Classification. The disposition of heating surface of the plain cylindrical shell boiler is defective on several counts. High volume-to-surface ratio makes it bulky. Large-diameter pressure vessels require thick walls for safety. The larger the volume of hot saturated water contained, the more violent the flashing action should the vessel ever be ruptured. A far better disposition of heating surface is had by using banks or nests of *tubes*. A tube may be defined as a hollow cylindrical vessel, with a length greatly exceeding its diameter, whose size is nominally its external diameter.* Boiler tubes are usually made of seamless steel. They range in size commonly from 2 in. to 6 in. diameter. The character and disposition of this tubular heating surface constitutes the principal difference between various types and makes of boilers.

Beginning with the simple cylindrical shell type boiler, heated with flames applied to the outer surface, many and varied types of boilers have been evolved. Some have characteristics which fit them for marine service, others for land service; some for power plants, others for heating, etc. For a comprehensive picture of the whole field, it is well to classify the group in logical order.

* Unlike *pipes* whose nominal size is their internal diameter.

CLASSIFICATION OF BOILERS

1. On the basis of the steam pressure.
   a. Low-pressure heating boilers, usually of cast iron construction throughout and made up in sections, somewhat akin to steam radiator practice. 5–10 psi. WSP.*
   b. Pressure boilers. Includes all other steam boilers. Mainly built up of cylindrical shapes, and fabricated of steel. The following items pertain to pressure boilers.
2. On the basis of the contents of the tubular heating surfaces.
   a. Fire tube, having flame and products of combustion in the tubes.
   b. Water tube, having the water and steam in the tubes, and the products of combustion outside.
3. On the basis of position of the furnace.
   a. Internally fired.
      The internally fired boiler has the furnace region completely surrounded by heating surface, as in the small portable vertical boilers, the Scotch boiler, or the locomotive boiler.
   b. Externally fired.
      All water-tube boilers are externally fired, and one of the fire-tube class, the horizontal return tubular, is externally fired.
4. On the basis of the shape and position of the tubular heating surface.
   a. The inclination of the tubes, as horizontal, inclined, or vertical.
   b. The form of the tubes—straight or bent.
5. On the basis of the drums.
   a. The number of drums, as single-drum, two-drum, or multiple-drum.
   b. The position of the drum with respect to the tubes, that is to say, across the tubes or parallel to the tubes, giving rise to the names cross drum and long drum. This distinction is not applied to bent tube boilers.
6. On the basis of the type of headers employed to connect the tubes to the drums of straight tube boilers.
   a. Sinuous, forged steel headers having a number of individual sections. Each section serves the tubes in a single vertical row, and there are as many sections side by side as the tube bank is wide.
   b. The box type riveted header, made of plate steel, and having sufficient area for the accommodation of all tube connections.

13–3. Fire-Tube Boiler. A fire-tube boiler is founded on a plate steel *shell*, usually cylindrical, since that shape best withstands internal bursting pressure. Holes are bored in the tube sheets, which are the ends of the cylinder, in such a manner that tubes may be passed through the shell and fastened tightly in the holes. These tubes are submerged in the water within the boiler, and hot-gas passes through them. In some boilers the surface of the tubes comprises the entire heating surface. Water circulation is random, but adequate, as a mass of rising steam keeps the water in turbulent motion. Figure 13–3 shows the construction of a fire-tube type known as the horizontal return tubular boiler, an externally fired type which has been very popular as

* WSP commonly abbreviates "Working Steam Pressure."

a source of steam for heating, industrial processes, and small amounts of steam power. It consists of a plain cylindrical shell with flat ends between which are supported a great many 3-in. or 4-in. iron or steel boiler tubes. The tubes do not entirely fill the shell, as a space must be left above them for the accumulation and storage of steam. The tube sheets above the tubes are braced against bulging by steel stay braces. The boiler is completed by the addition of proper taps for instruments, steam leads, feed-water pipe, and safety devices, and with the provision for some means of supporting it, such as brackets

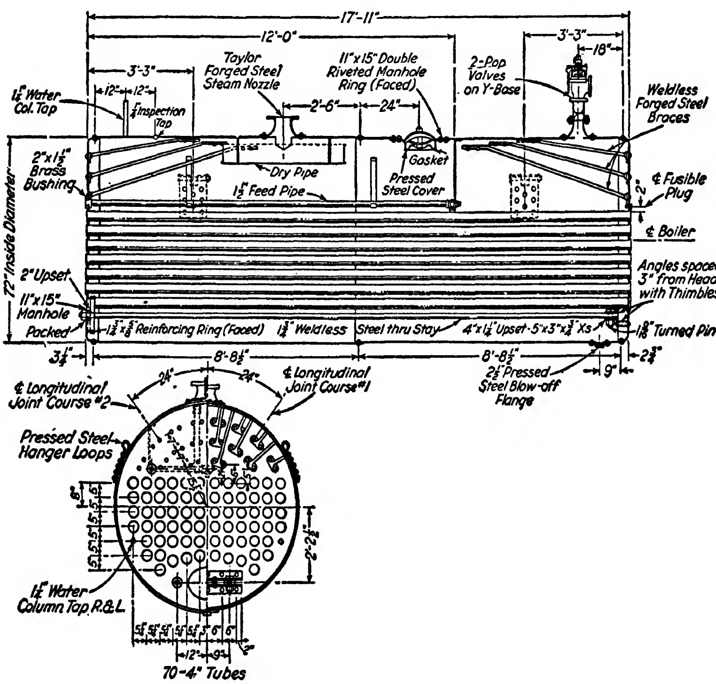


FIG. 13–3. Horizontal return tubular boiler. (Courtesy Wickes Boiler Co.)

or loops. The furnace is built below and external to the boiler shell, and both furnace and boiler are enclosed in a brick setting. The flames and hot gases play against the bottom of the shell, pass from the front to the back, and return to the front of the boiler through the tubes. The lower portion of the shell and the tubes form the heating surface.

Slightly different is the arrangement of the locomotive fire-tube boiler, shown in Figure 13–4, where the boiler is internally fired, and the flame and hot gases leaving the furnace region pass forward through the tubes. They emerge from the tubes into the smoke box, from whence they are blown to the atmosphere through a short stack, under the influence of a steam jet which receives steam from the exhaust of the locomotive cylinders. The heating surface consists of the tubes and the shell surrounding the furnace.

Similar to the locomotive boiler, but expressly designed for stationary service, is the fire-tube steel boiler, Figure 13–5. Here the construction of the

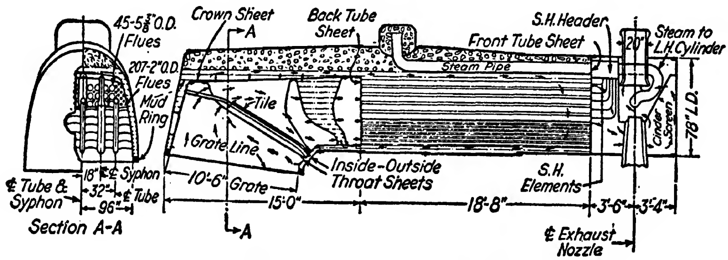


Fig. 13–4. Locomotive type boiler.

furnace-enclosing shell is clearly shown. That part of the boiler which extends down the side of the furnace is called a "water leg." It is restrained from bulging under the internal steam pressure by numerous *stay-bolts* which tie the two sheets together.

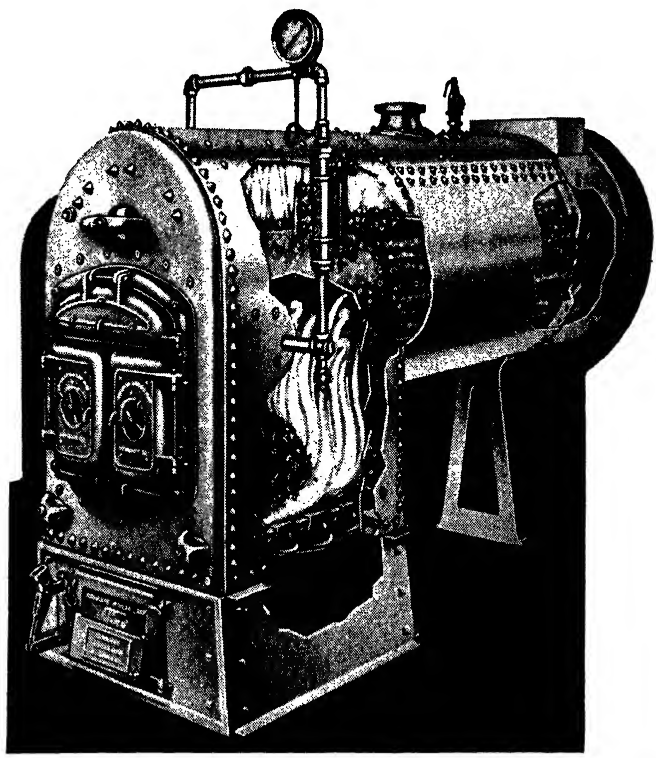


Fig. 13–5. Steel firebox boiler. (Courtesy Kewanee Boiler Corp.)

Another fire-tube type of some importance is the "Scotch marine," developed years ago in the Scottish shipyards for marine service. Nowadays it is

nearly superseded by water-tube marine boilers, but the principle has been developed into a small, compact, portable steam generating unit. Construction was stimulated by the extensive use of these "packaged" steam units by the military services during World War II. Were one of the lowest tubes of the horizontal return tubular boiler enlarged sufficiently to accommodate firing equipment in the internal furnace thus fashioned, we would have the basic feature of the Scotch marine boiler.

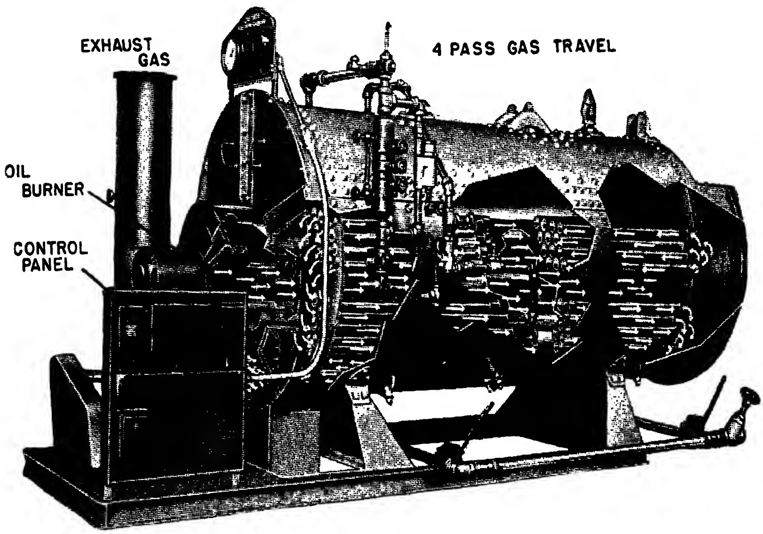


FIG. 13–6. The "miniature Scotch" type of unit steam generator. Features: Compactness, self-contained, oil-fired, mechanical draft, auxiliaries mounted on same base. Sometimes referred to as a "package" steam generator. (Courtesy Preferred Utilities Mfg. Corp.)

13–4. Water-Tube Boiler. A water-tube boiler is composed of drums and tubes, the tubes always being external to the drums, and a means of joining the tubes to the drums. The drums are used for storage of water and steam, and for connections of steam and water pipes. As they are not required to contain any tubular heating surface, they are much smaller in diameter than the shells of fire-tube boilers, and can be built for higher boiler pressures than are possible in fire-tube design. The heating surface is entirely in the tubes, the function of which is therefore simply to absorb the heat and generate the steam. The method of joining a tube to a drum or header is to insert that tube into a hole having the same diameter as the outside of the tube. The tube wall is forcibly expanded or rolled against the metal surrounding the hole, so as to make a tight joint. This process requires that the axis of the tube be perpendicular to the plane of the hole, or if the hole is bored in a curved surface, the axis of the tube must be radial to the curvature of the surface. If tubes are expanded directly into the drums, dispensing with an intervening

header, the only row of tubes that may be absolutely straight is that row that lies in the surface which joins the center lines of two parallel drums. All other rows of tubes must be bent at their ends so as to enter the drum surface radially. Some headerless boilers are built entirely with straight tubes by connecting the tubes to the flat ends of cylindrical drums, but the more common construction is that shown in Figures 13–7 and 13–8, where it can be clearly seen that the tubes are bent.

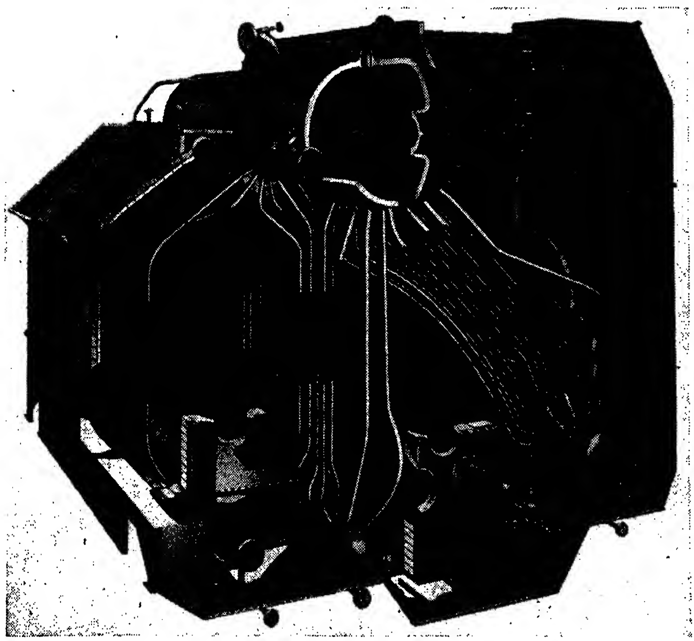


FIG. 13–7. Marine steam generator with separately fired superheater. (Courtesy Foster Wheeler Corp.)

Where tubes are left straight, *headers* will be provided to integrate the tubes into a coordinated *bank* of tubes. All tubes are the same length and are connected at either end to one or the other of two headers. Each header is connected to the drum by means of circulation tubes or sheet steel saddles. Box headers are thin hollow rectangular boxes of plate steel, bored to receive the tubes. Sectional headers consist of forged or cast steel sections, each accommodating a vertical group of tubes. As many sections are stacked side by side as are necessary to accommodate the desired number of tubes. Figure 13–9 shows the arrangement of surface in a straight tube box header boiler. As it is necessary to incline the tubes slightly to the horizontal in order to promote water circulation, the box type header must necessarily be inclined somewhat to the vertical, since the surface of the header must be

perpendicular to the tubes. Sectional forged steel headers may be exactly vertical, since an inclined surface may be forged at the opening for each tube so that the hole will be perpendicular to the axis of the tube.

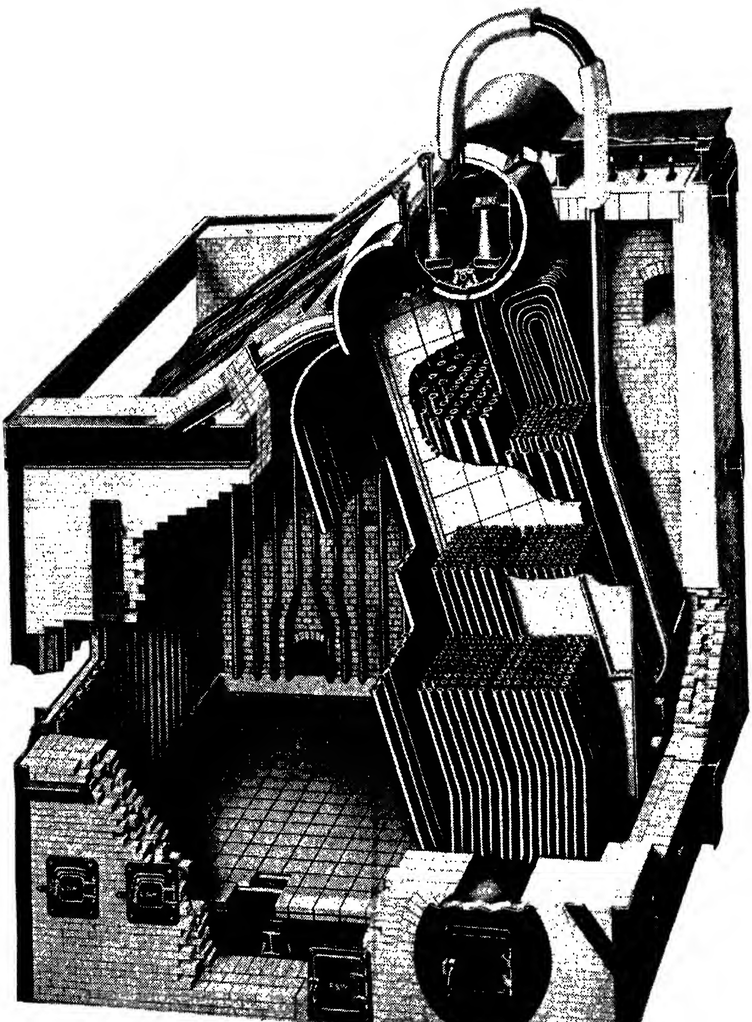


FIG. 13–8. Two-drum, bent-tube boiler. (Courtesy Babcock and Wilcox Co.)

The boiler element of the steam generator shown in Figure 13–10 would be classified as a sectional header, cross-drum boiler. Each section of the header end-connects eight groups of four tubes each, and an additional group of four circulation tubes. The sections are made with a slightly sinuous shape so that adjacent four-tube groups will be staggered enough to break up

stratification of the gases flowing through the tube bank. Tubes are inserted in the headers and the tightening tools used through the hand holes normally kept steam-tight by covers. Box headers have a similar hand-hole pattern.

The hydraulic circulation in these boilers arises from density differentials created by vapor-liquid mixtures existing in some parts of the circuit. Being

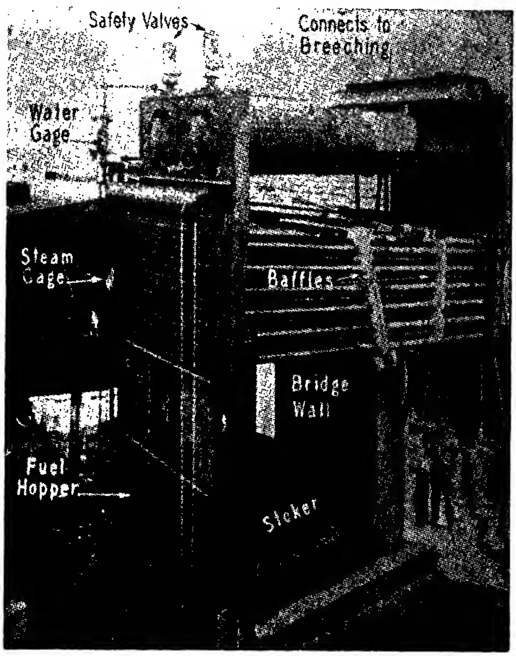


FIG. 13–9. Arrangement of longitudinal drum boiler. (Courtesy Edge Moor Iron Co.)

less dense than an equivalent column of liquid, the mixture is displaced toward the highest point of the tube. Bent tube designs usually have tubes with considerable vertical projection. Straight tube boilers could have horizontal tubes except for circulation, which requires that they be inclined at least 20° to the horizontal.

These boilers have *baffles*. A baffle is an object, usually a partition, placed for some specific purpose in the flow path of a fluid causing it to take some prearranged and circuitous path. Here, baffles direct the hot gas properly back and forth over the tubes so that the gas will give up its heat to the required degree, and will not short-circuit directly from the furnace to the stack. They are composed of refractory material similar to firebrick and will be found in longitudinal or transverse arrangement. Transverse baffling is made by building the baffle more or less perpendicular to the tubes. Longitudinal

baffles are usually precast and laid upon the tubes of the boiler, forming a baffle whose surface is parallel to the tubes. Reversals of the gas flow caused by baffles "pass" the gas back and forth over the heating surface. Such flow patterns are classified as single-pass, two-pass, three-pass, etc.

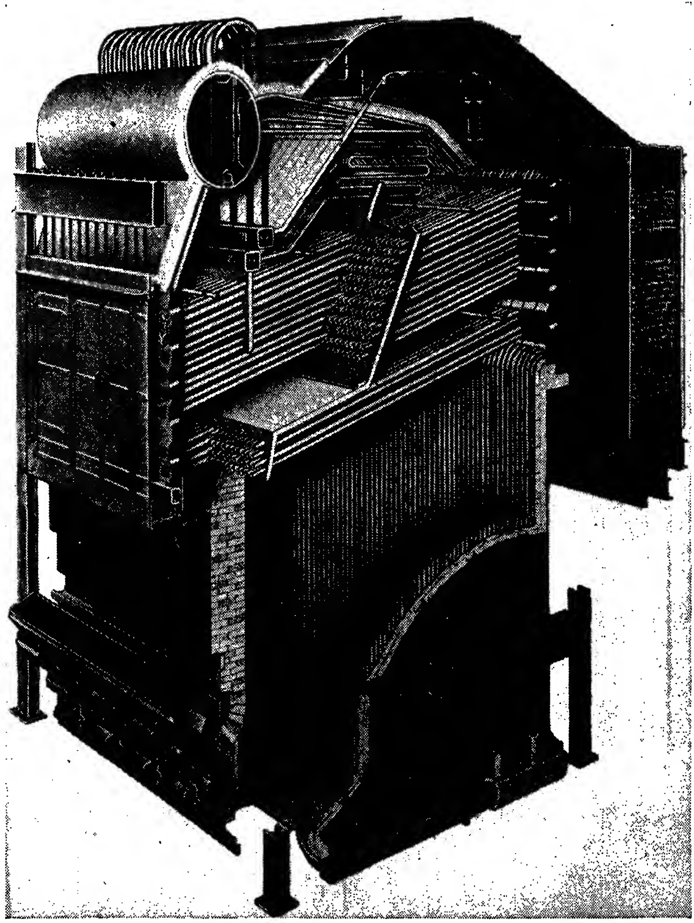


FIG. 13–10. Sectional-header, cross-drum boiler (with water walls, superheater and economizer). (Courtesy Springfield Boiler Co.)

13–5. Construction and Operation. The materials from which boilers are constructed are steel and iron. The low-pressure steam boilers so frequently employed in steam heating systems are generally made up of cast-iron sections. This material is entirely too weak and heavy for pressure boilers and rolled sheet iron and steel are employed in its place. The tubes are generally seamless iron or steel, and the drums are made of rolled steel sheets which are fabricated by welding or riveting, or both.

In operation, a boiler is supplied with heat from the combustion of a fuel, and with water from a pump known as the *boiler feed pump* which is capable of overcoming the pressure existing in the boiler. Upon entrance to the boiler, the water absorbs heat until it reaches its boiling temperature, and is then boiled off to form steam, which collects in the highest part of the boiler drum because of the difference in density of steam and water. Fresh feed water must be introduced to take the place of all steam generated and withdrawn from the boiler, so that the water level will be maintained at the proper position—midway up the drum of a water-tube boiler, and above the level of the tubes in a fire-tube boiler. Safety devices to protect against low water

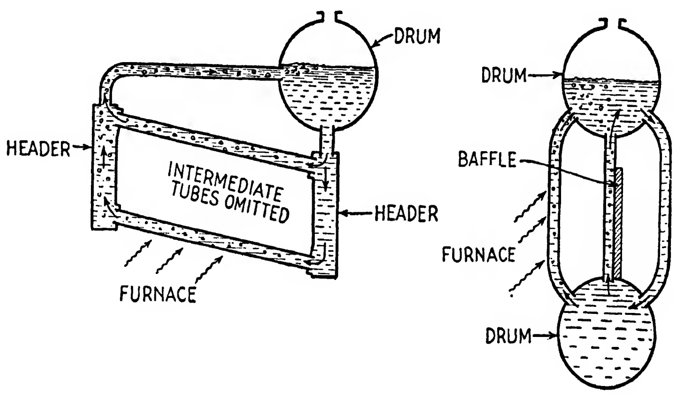
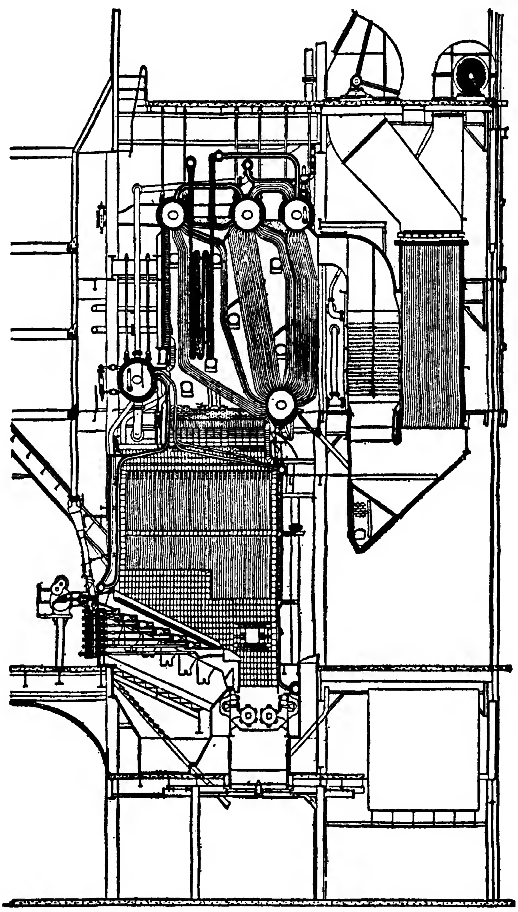


Fig. 13–11. Hydraulic circulation in water-tube boilers.

and high pressure must be provided. In addition to these safety devices, however, boilers are equipped with other accessories, such as dry pipes to filter the entrained droplets of water from the steam and deliver dry steam at the steam nozzle, water level gages, drain and blowdown connections, superheaters, soot blowers, water walls, and steam pressure gages. Operation of the whole steam generating unit may also involve combustion equipment, draft fans, chimney, breechings, economizer, and air preheater. Figure 13–12 shows a sectional drawing of a large power boiler in which many of these auxiliaries are pictured. An air preheater and economizer, fans, water wall, stoker, and superheater may be noted. Many of these auxiliaries may be seen if the other illustrations are carefully examined.

Given service conditions can usually be met equally well by several boilers of different design. However, in spite of the wide variations in design, there are certain requirements which are fundamental to all water-tube boilers. All boilers deserving consideration should meet these requirements, although, of course, not all the favorable points of different designs are covered. But it is nearly certain that if the boiler does not conform to them, operating difficulties will occur early in its life. First, there are the conditions governing the

behavior of the water within the boiler. Most important of these is good water circulation. The disengagement surface where the steam breaks through the surface of the water in the drum should be as large as possible. Priming * may result from restricted disengagement surface. In order to control



American Engineering Co.

FIG. 13–12. Cross-section of steam-generating unit, Delray No. 3 Station, Detroit Edison.

impurities which will be precipitated from the feed water, it should discharge into the drum at a point where the circulation will deposit the precipitate in a settling chamber called the mud drum.

The path of the gases through the boiler should be so baffled and directed over the tubes that they give up heat to the required degree. This required degree is less when auxiliary heat transfer surface in economizer or air preheater is provided than when it is absent. Certain features of a boiler may

* Generation of excessively wet steam.

result in undetermined thermal stresses being set up, such as the discharge of cold feed water against the boiler shell setting up contraction stresses. Joints and seams should be well protected from the flames, and burners should never be set so the flames play directly against the surfaces. To provide for intelligent and safe operation of the boiler a full complement of leads, gages, and safety devices are provided. These include blow off, steam lead, feed-water lead, water gage, pressure gage, safety valve, and fusable plugs. Boilers must be accessible for inspection externally and internally. Not only the owner, but insurance inspectors need to examine a pressure boiler at intervals to determine its condition and safety.

13–6. Capacity. Steam generating units are installed to meet some need for steam at a predetermined physical state. An obvious way to specify capacity is therefore to name the steam pressure, quality or temperature, feed-water temperature, and quantity that shall be produced per hour. This *specific capacity* is often employed as a rating system for large steam generators, the design of which is specially tailored to the purchaser's requirements. In the smaller capacities, identical boilers of a manufacturer's "line" may be offered to customers having somewhat different steam raising conditions. Then a more flexible rating system is desirable. Sometimes "*K B.t.u.*" is used, meaning Kilo B.t.u.'s transferable per hour. More often these boilers are rated by a hypothetical "boiler horsepower."

The term *boiler horsepower* was defined in 1876. At that time the average engine would operate on 30 lbs. of steam per hp. per hr., and a boiler horsepower was defined as the capacity to generate steam at that rate.* The average rate of evaporation, then, was 3 lbs. of steam per sq. ft. of heating surface per hour in a water-tube boiler, so 10 sq. ft. of heating surface was accepted as manufacturer's rating for a boiler horsepower. (12 sq. ft. for a fire-tube type.) The 10:30 ratio of heating surface to steam rate became obsolete when steam rates were reduced to less than 10 lbs. per hp. hr. through improvements in steam turbines and engines, and evaporative rates were increased from 3 to 10 or more lbs. per sq. ft. per hr. by improvements in boiler design and firing.

Yet the 10 sq. ft. per boiler horsepower persisted, so that now capacity can be expressed in terms of the manufacturer's horsepower rating only by misuse of the term "per cent rating." A per cent rating of more than 100% is usually construed as an overload, but not necessarily so in the boiler field, where per cent rating is simply the ratio of developed heat transfer to an arbitrarily selected rating system. When the significance of the term "per

* To standardize the heat so involved 33,500 B.t.u. is taken as the transferred heat of a boiler horsepower, this being equivalent to the 34.5 lbs. of steam generated at 212° F, from feed water at 212° F, that was adopted as a standard "developed boiler horsepower."

cent rating" is understood, there is no disadvantage to the use of boiler horsepower actually developed, as a boiler horsepower = 33,500 B.t.u. per hour transferred to the water.

Curved elements like tubes possess two different areas, the inside and outside surfaces. Which is the official "heating surface"? It is considered to be the one in contact with the high temperature fluid, not only in the boiler field, but generally throughout surface heat-transfer equipment. The heating surface of fire-tube boilers is therefore the interior tube surface; whereas it is the exterior surface for water tubes.

Boiler horsepower is sometimes used to describe the capacity of steam generators which have superheating as well as evaporating surface. As these surfaces transfer at different rates, this practice is confusing, and should be abandoned. The term "boiler horsepower," if employed, should be restricted to evaporative heat-transfer surfaces. Low-pressure steam-heating boilers are customarily rated in terms of radiator capacity they will serve, i.e., square feet of equivalent direct radiation (EDR).

Example 1: The heating surface of a boiler like Figure 2–4 consists of 50 tubes each 8 ft. long, 4 in. in diameter, and $\frac{3}{16}$ in. thick. The interior shell surface exposed to the furnace is estimated to be 40 sq. ft. What is the rated horsepower?

This is a fire-tube boiler, for which a horsepower is 12 sq. ft. of interior tube surface. The heating surface is computed as that of 50 cylinders, each $4 - (2 \times \frac{3}{16}) = 3.625$ in. diameter and 8 ft. long.

$$\text{Tube area} = 50 \times 8 \left(\frac{3.625\pi}{12}\right) = 380 \text{ sq. ft.}$$

$$\text{Rated horsepower} = \frac{380 + 40}{12} = 35 \text{ bo. hp.}$$

Example 2: Steaming conditions of a boiler are such that 1095 B.t.u. must be absorbed by each pound of water being converted to steam. What developed boiler horsepower is represented by a steam generation rate of 1525 lbs. per hr.?

$$\text{Heat transfer per hour} = 1525 \times 1095 = 1{,}672{,}000 \text{ B.t.u.}$$

$$\text{Developed horsepower} = \frac{1{,}672{,}000}{33{,}500} = 50 \text{ bo. hp.}$$

Example 3: Assume that the data of Examples 1 and 2 pertain to the same boiler. What is the operating rating?

This boiler is operating at $\frac{50}{35}$ or 142.5 "per cent rating."

Example 4: A water-tube boiler with 8000 sq. ft. heating surface is operated so as to produce 72,000 lbs. of steam per hr. dry and saturated at 200 psi. from 200° F feed water. Express the operating conditions as (a) K B.t.u., (b) boiler horsepower, (c) per cent rating.

Enthalpy of the steam (from tables) = 1198.7 B.t.u. per lb..
Enthalpy of feed water = 168.0 B.t.u. per lb.
Heat transferred per hour = 72,000(1198.7 − 168.0) = 74,200,000 B.t.u.

a. Heat rating = 74,200,000/1000 = 74,200 K B.t.u.
b. Boiler horsepower = 74,200,000/33,500 = 2210 bo. hp.
c. Rated horsepower = 8000/10 = 800 hp.
Per cent rating = 2210/800 = 276%.

Example 5: Using the data of Example 4, express the capacity as equivalent evaporation "from and at" 212° F.

To evaporate steam from feed water at 212° into dry saturated steam at 212° (atmospheric pressure) requires only h_{fg} to be added. At 14.7 psi., h_{fg} = 970.3 B.t.u. per lb. From Example 4 we see that 74,200,000 B.t.u. were transferred per hr. This would have evaporated 74,200,000/970.3 = 76,400 lbs. "from and at" 212°. This is the *equivalent evaporation.*

13–7. Superheater. The advantage of having a high initial temperature for any power cycle was demonstrated in prior chapters. Obtaining these high temperatures in steam, by using higher boiler pressures, is attended by these disadvantages. (1) High pressures require stronger, more costly boilers; (2) Up to about 650 psi. the saturation temperature of steam increases nicely along with the pressure but thereafter begins to lag and, subsequently, major pressure increases bring only minor temperature rises.* On the other hand, while superheating is capable of high temperatures at moderate pressures, only a small part of the heat is raised to the high temperature. After the prime mover has used the heat of superheat the remainder of the available energy exists at saturation temperature. So neither high pressure nor superheat is an unmixed blessing in the attainment of highpower cycle efficiency. Usually a combination of moderately high pressure and superheat is used where good efficiencies are being sought.

The superheater is heat-transfer surface, tubular in character, arranged to receive saturated steam at the inlet of the tubes and deliver superheated steam steadily at the outlet. Heat is received from combustion of fuels by direct radiation to the surface of the superheater tubes, by convection of hot furnace gases made to flow over the tube surfaces, or both. Superheater tubes are made small in diameter to promote maximum contact of the steam with the hot tube wall. Usually the superheater is placed within the boiler setting and thus uses heat from the boiler furnace, but separately set and fired superheaters have occasionally been employed. A separately fired superheater is illustrated in Figure 13–7, integral types in 13–8, 13–10, and 13–12. Superheaters are classified as *radiant* or *convection* types, depending on their location in the setting. Heat transfer rates from dry gas to dry steam are comparatively low, hence superheater tubes must be quite long if the steam passing through is to gain a substantial superheat. Such long lengths can be accommodated in the boiler setting only by "folding" the tubes several times. Figure 13–10 shows this feature clearly.

* Although the temperature of saturated steam is almost 500° F at 650 psi., only about 200° F more can be obtained by an additional 2500 psi. See steam table data.

Since the maximum steam temperatures now employed are close to the working limit of metals employed, close control over the superheat is essential in the steam generator of advanced design. Superheat may be controlled in a number of ways, two of which are:

1. Follow the superheater with a water spray desuperheater operated by a temperature regulator.
2. By-pass gas. This method is employed for regulating convection superheaters. It is necessary to employ an oversize superheater, one which will give the required superheat at the lowest specified load. At higher loads the by-pass is opened, allowing some of the gas to flow around rather than over the superheater. Although the gas passing the superheater is hotter, its volume is less, hence the compensation.

13–8. Water Wall. Radiant heat is more speedily transferred to the water than heat that is passed by convection. All boiler tubes, which are directly exposed to the region of combustion, transfer heat at much higher rates than elsewhere. High-capacity boilers often have considerable extra heating surface arranged around the furnace walls to gain the advantages of radiant heat absorption. These surfaces are called water walls. In large steam generators they may account for most of the steam raising capacity. Water walls are especially in evidence in Figures 13–8 and 13–10. When a fuel like gas or pulverized coal is used, little excess air is required for complete combustion. The combustion temperatures are likely to be exceedingly high unless large quantities of energy are radiated away from the combustion zone immediately. This is another factor favoring use of water walls.

13–9. Boiler Accessories. Safe operation of a fired pressure vessel, such as a boiler, implies, on the part of the operator, a continuous knowledge of the water level and the steam pressure. This is provided by the *water column* and the steam *pressure gage.* Protection against excessive steam pressure is afforded by a *safety valve,* against low water by such automatic alarms as fusible plugs, whistles, etc.

The water column shown in Figure 13–13 consists of a chamber connected above and below the normal water line so that the water level in the boiler shell or drum is duplicated in the column. There quick-acting cocks are mounted on the column so that the general location of the water level can be tested by successively opening the cocks and noting whether steam or water is blown forth. A water column installed will be seen in Figure 13–5. Note also the steam gage connection. A continuous visible water level is shown in the transparent tubular glass gage attached to the column.

As the connections to the gage glass might become clogged with sediment, causing a false level to be shown, it is provided with a means for quickly blowing the water from it, after which the operator can see it fill to the true level.

Water columns are often furnished internally with a float which will admit steam to an alarm whistle when the water level reaches the bottom of the column. In some cases the float also causes the alarm to sound if the water level creeps unduly high (as it might if feed-water regulators became dearranged).

Safety valves which were once commonly loaded by a weight on a lever are now spring-loaded. The valve is held tightly against the seat by a heavy calibrated spring. Whenever the steam pressure against the valve reaches the intensity for which the valve is "set," steam and spring pressures are in equilibrium and any slight steam pressure increase will lift the valve slightly. Immediately steam can bear against a larger surface and the valve "pops" wide open and remains so until a predetermined pressure drop (2% to 4% of WSP) has occurred due to escape of steam to the atmosphere. This action is designed to keep the valve from floating in a partially open position when the rate of steam generation is only slightly excessive. Steam escaping through a partially opened valve would soon cut and channel the valve and seat, making tight closure subsequently impossible.

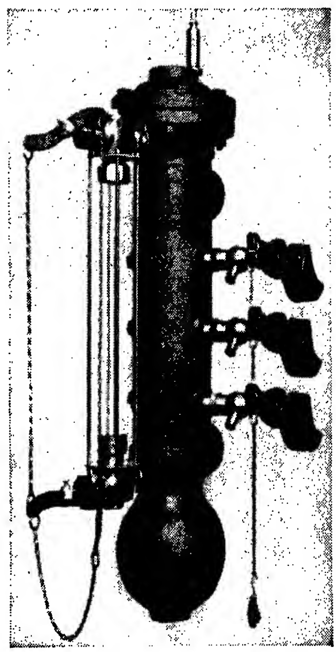


Fig. 13–13. Boiler water column. (Courtesy Wright-Austin Co.)

The term "boiler trim" generally refers to the collection of fittings and gages which are mounted on the boiler after erection. It includes steam gage, water column, feed valves, safety valves, blow-off (drainage) valve.

13–10. Furnace. The furnace is a chamber for combustion. In addition it provides support and partial enclosure of the firing equipment. It surrounds the high temperature region, confining and isolating the combustion reaction so that it can be controlled. The shape and size of a furnace are dependent on:

1. Type of fuel (gas, oil, coal).
2. Firing equipment used.
3. Type of boiler.

It is frequently of rectangular, box-like form, but can take almost any needed shape, even cylindrical, as in some internal furnace fire-tube designs. Apart from having a shape that will receive the combustion from the firing equipment and deploy the heat energy to the boiler surfaces, the furnace must be satisfactory on these two counts. (1) The walls must be adequate to stand

the interior temperature without rapid deterioration, and to resist heat transfer through them to a satisfactory degree. (2) The internal volume must be large enough to hold the combustion process in the hot furnace long enough for it to become complete.

Furnaces can be classified according to their wall construction, as (1) solid, (2) air-cooled refractory, or (3) water cooled. Interior walls must be faced with a material capable of standing temperatures of 1500–2500° F. Materials which will resist change of shape, weight, or physical properties at

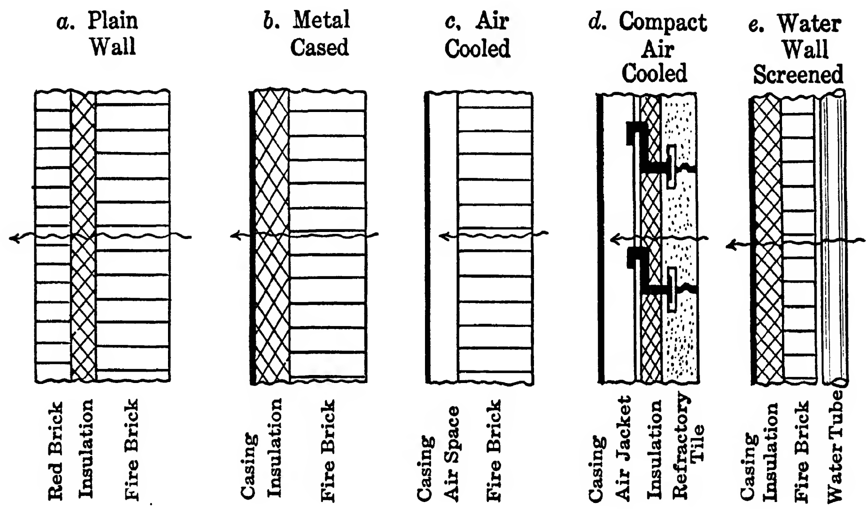


FIG. 13–14. Furnace wall sections. (Arrows give direction of heat flow.)

high temperatures are known as refractories. The materials which are chiefly used for refractories are fire clay, silica, kaolin, diaspore, alumina, and certain products of the electric furnace, such as silicon carbide. Refractories are used most often in the form of bricks.

A refractory brick is built primarily to withstand temperature. It is not a good insulator. Indeed, the most refractory bricks usually have the highest thermal conductivities. It is important for the refractory brick to have high resistance to erosion by ash-laden gases and to the fluxing action of molten slag. It should not spall badly under rapid temperature changes, and its structural strength should hold up well as fusion temperature is approached. Fire clay bricks are made from certain clays, including a plastic clay which binds the others into brick form. The firing in the kiln is carried out at a temperature such that the brick is partly vitrified. Progress in the art of combustion of fuels in furnaces has advanced the service requirements of refractory brick, sometimes to the point where they are so severe that a refractory superior to fire clay is needed. High alumina brick containing from 50% to 80% alumina, and silicon carbide are typical of these superrefractories.

Of course fire clay bricks are preferred wherever they give satisfactory service because they are lowest in cost of all the refractory bricks. The standard size of fire brick is 9 in. $\times$ $4\frac{1}{2}$ in. $\times$ $2\frac{1}{2}$ in.

Some typical walls of solid construction are shown in Figure 13–14. The flow of heat energy under the influence of temperature difference is akin to the case of heat transfer illustrated in Figure 4–2. Heat flow through the wall creates a temperature drop through the refractory layer, whose thickness must be great enough so that the temperature of the next layer will be within the ability of that material to stand. The outside wall temperature will have to exceed ambient atmospheric temperature sufficiently to discharge the heat flow into the surroundings.

Example 1: A square foot of brick wall 80° F hotter than the surroundings will discharge about 150 B.t.u. per hr. Assuming this condition for the wall section *a* (Figure 13–14), how thick will the refractory layer have to be if the insulating layer should not exceed 1700° F? Conductivity, K, of the refractory material (fire brick) is 0.9 B.t.u. per sq. ft. per hr. per deg. per ft. Inside furnace wall temperature 2000° F.

The conductance of the refractory layer must satisfy the equation $Q = UA\Delta T$, in which $Q = 150$, $A = 1$, $\Delta T = 2000 - 1700 = 300°$ F.

Then,

$$U = \tfrac{150}{300} = .50 \text{ B.t.u. per hr. per sq. ft. per deg.}$$

Also,

$$U = \frac{K}{d} = \frac{0.9}{d}.$$

Solving for d,

$$d = \frac{0.9}{0.50} = 1.8 \text{ ft.}$$

The foregoing example shows that solid walls are liable to be very thick if they have to cope with high furnace temperatures. Two methods for reducing bulk and weight of furnace walls are represented by the other wall classifications. An air-cooled wall is a refractory layer backed by an air jacket through which a continuous stream of air is circulated. The refractory in contact with the air is very hot and heats the air considerably, but the heat is not wasted, for the stream of cooling air is really the combustion air on its way to the firing equipment. The third classification is the water wall, which has already been described. In this type the refractories are protected from the intense furnace temperature by a screen of water-filled tubes so their initial temperature is considerably lower than for a plain solid wall.

The furnace must have a volume adequate to the kind of combustion used. If the volume is insufficient, gases will pass prematurely into the tube bank and be chilled enough to arrest the combustion reaction. Rapidity of combustion depends much on the relative motion between fuel and air—on turbulence. Air rushing through a bed of lump coal resting on grates provides maximum relative motion and rapid combustion. Air and pulverized coal

moving into a furnace from a burner have only the turbulence which the burner design imparts, hence require far more furnace volume for complete combustion than is required for lump coal. So the proper volume for a furnace depends on kind of fuel and how it is fired, as well as the quantity of fuel. Some of the arbitrary methods which have appeared are (1) to allow a certain volume per rated horsepower, (2) to allow a certain volume per square foot of grate area, (3) to allow a certain "heat liberation" per cubic foot of furnace volume. The heat liberation method is somewhat more rational than the others, but requires empirical data to activate it. *Heat liberation* is the number of B.t.u. produced per hour per cubic foot of furnace volume. To determine furnace volume by this method requires that the potential heat input in the fuel expected to be burned per hour be divided by some heat liberation rate, previously found to be adequate. These rates are of the order of 30,000 B.t.u. per cu. ft. per hr. for lump coal, 25,000 to 40,000 B.t.u. for fluid fuels, and from 12,000 to 20,000 B.t.u. for pulverized coal (high values are for furnaces with water walls and for coals of comparatively high ash melting temperature).

Example 2: Estimate the required furnace volume to burn 20 gals. light fuel oil per hour. Assumed heat liberation rate, 30,000 B.t.u. per cu. ft. per hr.

Light furnace oil has a heating value of approximately 18,500 B.t.u. per lb. and a specific gravity of .85. Twenty gallons weigh $20 \times 8.33 \times .85 = 142$ lbs.

$$\text{Furnace volume} = \frac{142 \times 18{,}500}{30{,}000} = 88 \text{ cu. ft.}$$

13–11. Firing Equipment. The functional location of this item is shown in Figure 13–1. It is attached to or sets in the furnace. It is the gateway through which the fuel and air enter the combustion zone. But this equipment is more than just a channel for the inflow of the raw materials of combustion. It must achieve the proper air-fuel ratio and obtain good mixing of air with fuel. Also it sometimes provides support of the fuel during combustion (lump fuel).

Firing equipment becomes a so-called "burner" when the fuel is fluid. Solid fuel is supported on grates while it burns. If the fuel feed and manipulation of the fuel bed are mechanical rather than manual, the equipment is known as a stoker.

Burners are commonly provided for liquid and gaseous fuels, but pulverized coal, when mixed with a certain amount of air which lubricates it, flows much as a fluid, and is fired by burners. Burner designs for gaseous fuels are the simplest because gas and air are easily mixed and a gas burner may consist of little other than a means to subdivide the total flow of the fuel into a large number of small jets which will present a large surface exposure to the combining oxygen. A liquid fuel will approach the efficiency of gas firing

after it is sufficiently atomized or vaporized so as to present a large surface for combustion. This can be done by wicks, by contact with heated metal as in wickless kerosene burners, by the use of high-pressure sprays through very fine holes, or by the use of steam jets. Fuel oil burners of large capacity accomplish an atomization of the oil mechanically by giving the oil a whirl in the burner tip, and discharging it into the furnace through a small orifice. When steam is used, the oil is broken up by a continuous discharge of steam under

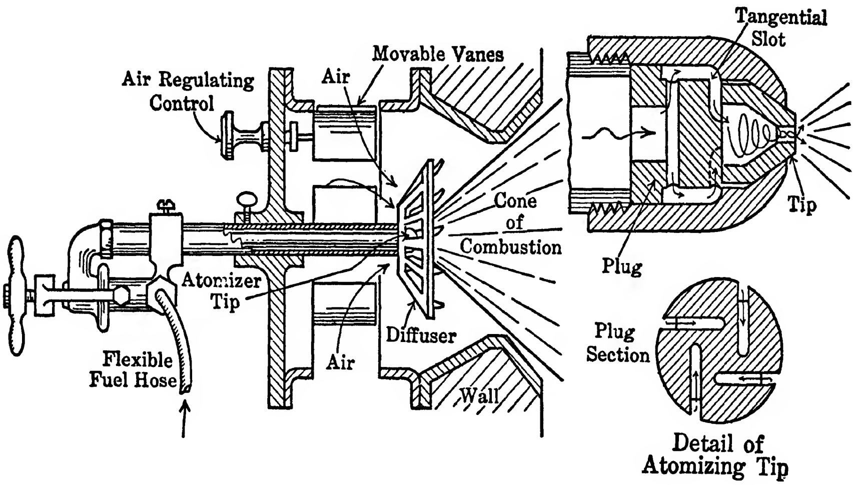


FIG. 13–15. Mechanical atomizing oil burner. Quantity of oil sprayed may be adjusted within moderate limits by varying the oil pressure supplied. However, a set of plugs and tips with varying oil passages will permit a wide range of quantity sprayed, since the burner tube may be withdrawn and these elements changed with very little trouble. If constant attendance is not available and automatic control is installed, special types of wide-range burners must be substituted for the above design.

pressure within the burner and at the tip outlet. Preparation of the oil for use in the burner may include filtering, heating, and pumping.

Pulverized coal offers much more of a problem in burner design than do oil and gas, even though the coal is pulverized to such an extremely small size that 90% of it will pass through a screen having 100 openings to the inch. The pulverized coal is floated or air-borne to the burner on a stream of air amounting to some 10% or 20% of the total combustion requirements. This is called primary air. The remainder of the air needed, called secondary air, is admitted directly to the burner, the function of which then becomes one of properly proportioning the fuel and air, and thoroughly mixing them.

Figure 13–16 shows the primary and secondary air inlets into the short flame burner. The rate of combustion is regulated externally by the rate of feed of coal to the burner. Proper proportioning of air is accomplished by a damper in the secondary air passage. Air vanes cause rotation of the secondary air moving into the furnace. A fan type spreader sets the primary air

and coal whirling in the opposite direction. With this turbulence, and in the presence of a furnace temperature higher than that required for ignition, the flame will propagate itself back into the incoming air-fuel stream at from 15 to 40 ft. per sec. The velocity of the mixture leaving the burner must exceed this in order to prevent flarebacks.

Grates are bars which are arranged to support solid fuel for combustion. The purpose of a grate is not only to support the fuel bed, but also to allow air to pass into it evenly. Service conditions of grates call for considerable structural strength, resistance to temperature and oxidation, and a shape serrated or rough edged, so as to leave air openings between adjacent grate bars. Furthermore, since ashes are generally dumped through the grate bar system, a means for increasing the spacing between bars so as to pass ash between them must be provided in such a way that motion of a shaking crank or lever will clear ash over the entire grate surface. Because of its cheapness and its resistance to oxidation, cast iron has been the material most used for the construction of grates.

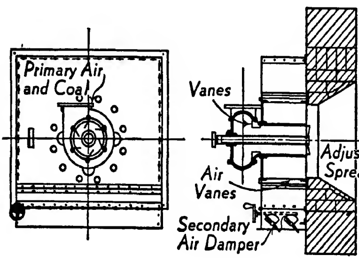


FIG. 13-16. Circular burner. (Courtesy Riley Stoker Corp.)

Modern *stokers* may be classified as overfeed, underfeed, and conveyor. The sprinkler or spreader stoker is a type of overfeed, having horizontal grates. A rotor is driven at high speed and so located that any coal dropped on it will be thrown onto the grates. A feeder delivers the coal at the proper rate from the hopper into the range of action of the rotor blades. Some sprinkler stokers have a pneumatic or steam jet delivery of the coal to the grates. The coal should either be screenings or be crushed to small size. The fine particles are mostly burned in suspension, while the heavier pieces fall to the grates and burn there.

Stokers most in use nowadays are the conveyor and underfeed types. A chain grate stoker (one type of conveyor) is a broad endless belt composed of short connected links. This belt passes over two sprockets so that it makes a flat upper surface upon which coal can rest. One of the sprockets is power driven, so that there is a very slow motion of the grate in a direction which will drag coal from the bottom of the hopper, carry it into the furnace, and finally dump the ash into a hopper. The rate of combustion is varied by simultaneous control of air pressure, velocity of the grates, and thickness of the fuel bed. Heat from the incandescent zone is radiated to the ignition arch and reflected back to the green fuel bed at its entry to the furnace. The volatile matter that is driven off is mixed with air admitted through and over the green fuel bed, and, being confined to the arch, is passed through the hot-

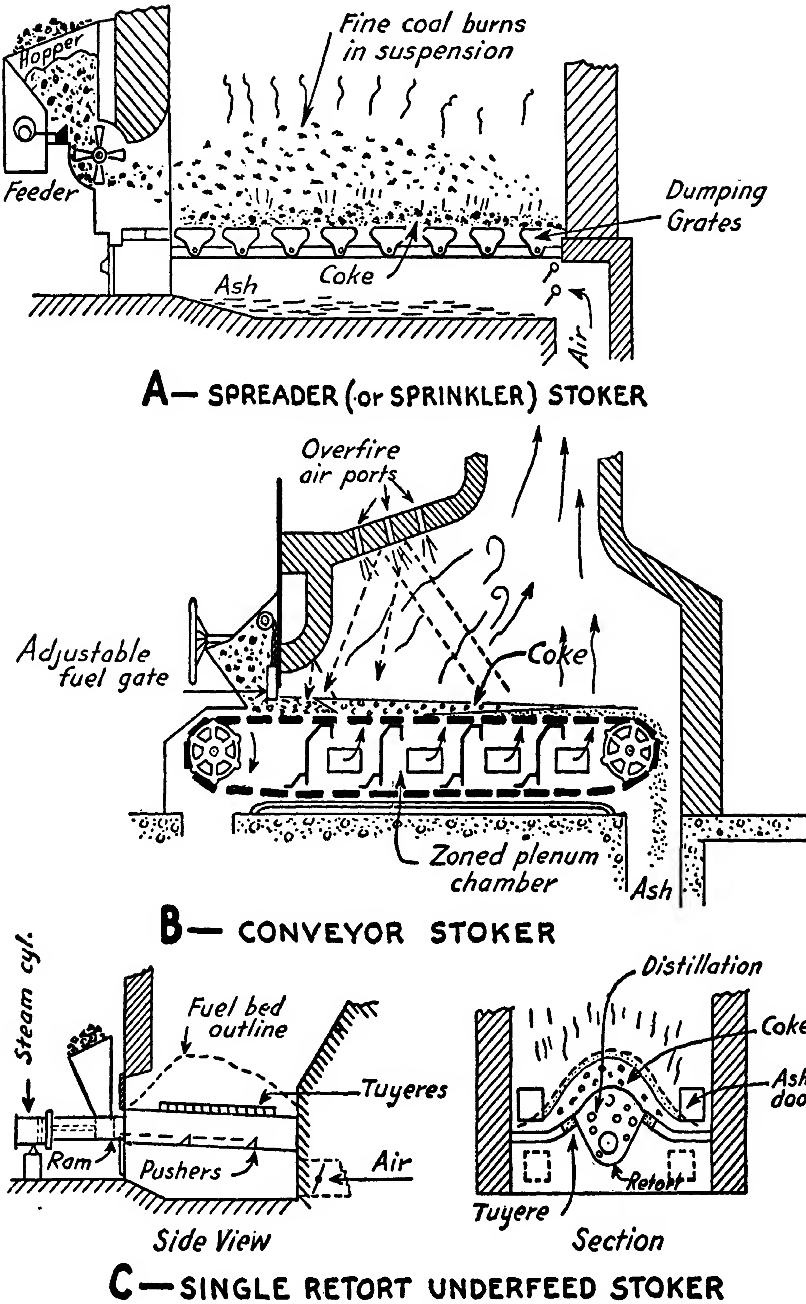


Fig. 13-17. Principles of stoker combustion.

test zone of the furnace. The carbon left behind then reaches the ignition point and burns, giving up heat, part of which is radiated back to the incoming green coal. There is no means of breaking up a crust formed by a coal which will soften and fuse together, but the chain grate stoker will burn coals that would be too fine or easily packed for use in an underfeed stoker, as well as coals that clinker badly.

The efficient combustion of certain classes of coal requires that the coal be gasified under conditions which will prevent it from fusing into an air-tight crust. A coal that has a tendency to fuse together into a solid cake of coke must be burned in an underfeed stoker, in which the fresh coal is fed to the fire zone by being pushed up from underneath. This type of feed creates a heaving action at the fire line, which tends to prevent the formation of an air-blanketing crust. An underfeed stoker consists of a retort, usually trough-shaped in industrial stokers, but resembling a pot in the domestic type. The coal is forced into this retort by an endless screw or ram in such a way as to accomplish the feed action outlined above. Air is supplied at the sides of the retort under pressure sufficient to enable it to penetrate the zone of incandescence. Heat from an incandescent zone of burning carbon is radiated and conducted downward into the green fuel which is being crowded up to the ignition line from below. The volatile matter which is given off is mixed with air supplied from tuyères, also below the ignition line. The inflammable mixture then passes directly through the hottest zone of the furnace—the incandescent fuel bed—insuring that all parts of it reach the ignition point. The ever-upward motion of the fuel keeps the bed well broken up, allowing free passage of gases and oxygen to all parts of it.

The single retort underfeed stoker is well adapted to a wide range of coals and to all sizes up to 1000 boiler horsepower. The size of a retort is limited by the ability of the air to penetrate from the tuyères to the zone of burning. Large underfeed stokers cannot be built with single large retorts, for this reason; rather, they must be built with a number of parallel retorts. However, the multiple retort underfeed stoker does not resemble a number of single retort stokers placed side by side. Instead, the retorts are sloped downward slightly, and the coal emerges from the retort on to an overfeed section, where the residual carbon is burned.

13–12. Pulverized Coal. One method of firing coal is to grind it to a dust and use a burner for the firing equipment. This fine grinding exposes an enormously greater fuel surface to the action of oxygen during the process of combustion. Very little excess air is then needed to secure complete combustion. That is the principal advantage of this system of firing coal and it results in considerable saving of fuel because the 30% to 80% excess air that must be supplied to burn lump coal completely carries off much heat in the form of hot chimney gas. The combustion of pulverized coal is readily adapt-

able to automatic control, responding rapidly to control manipulation. There is but little loss analogous to the banking loss of stoker fires. Pulverized coal firing utilizes preheated air to good advantage and without limitation of the preheated air temperature. Boiler room cleanliness is another asset of the pulverized coal installation.

These advantages are not secured without some undesirable features. Pulverized coal brings with it additional costs and complications of equipment. Much of the fine flocculent ash would pass up the stack and spread

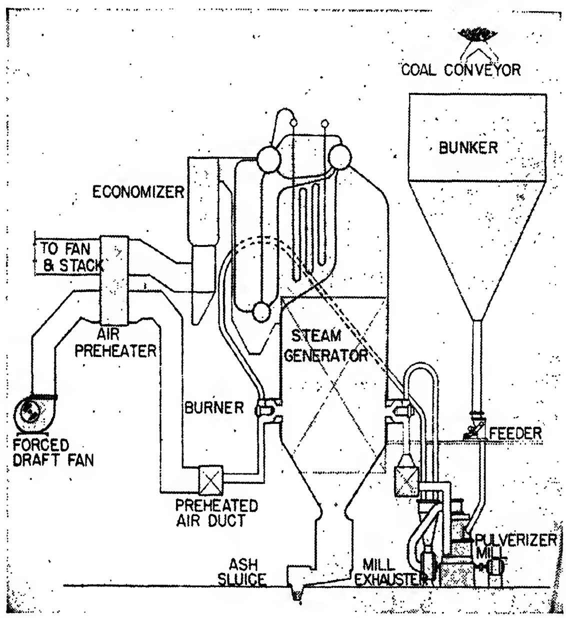


FIG. 13–18. Pulverized coal system.

itself upon the surroundings were not strenuous efforts made to precipitate most of it. More power is required than for stokers. Low excess air and high capacity raise the temperature of the furnace to where it must be water cooled to avoid excessive refractory maintenance.

There are two pulverized coal systems. They are called the central system (bin system) and the unit system. A central pulverizing system employs a limited number of larger capacity pulverizers at a central preparation point to prepare coal for all the burners.

The unit system, so called from the fact that each burner, or burner group, and pulverizer constitute a unit, is the one generally used. Crushed coal is fed to the pulverizing mill at a variable rate governed by the combustion requirements of the boiler and furnace. Preheated primary air is admitted to the mill and is the transport air which carries the coal through the short delivery pipe to the burner.

The pulverizing mills may be classified as (1) impact mills in which a series of swinging hammers or falling balls pulverize the coal, (2) roller mills in which grinding is done by crushing between rollers or balls and a race, (3) chopping or attrition mills.

13–13. Performance. A considerable part of this chapter has been devoted to matters of general import in the field of steam generation, to features of different boiler types, and to matters which are to be weighed and analyzed in the design office or by the prospective purchaser. When the scene shifts to a boiler room where steam generating equipment is in service, interest shifts to the performance of that particular boiler. Steam generator performance is the aggregate of several indices, among which *availability* and *efficiency* are outstanding.

Availability, which means availability for service, is sometimes stated as the percentage of the time that a steam generating unit is either in service or could be put into service if desired. Outages for repairs, inspection, cleaning, and modification will cut into availability, but central station steam generator records often show 85% or better availability for service. Boiler outage may enforce a power plant shutdown. This may close down an industry, or excessively inconvenience large numbers of public service customers. Availability is possibly of more importance than efficiency to the plant owner, especially if his steaming capacity is concentrated in one unit. Maximum availability, however, is obtained by features which usually add to the cost and bulk of a steam plant. Obtaining maximum availability from an existing steam unit has to do with careful attention to firing conditions, keeping surfaces clean with soot blowers, not having to operate at excessively high ratings, etc.

The cost of the heat energy represented by steam is likewise a matter of concern to the operator. This varies inversely with the *thermal efficiency* of the generating unit. By thermal efficiency is meant the measure of ability of a steam generator to transfer the heat given it by the furnace to the water and steam. This will sometimes include superheater performance. Boiler and furnace are so much a unit nowadays that "boiler, superheater, and furnace" efficiency is more important than boiler efficiency. In fact, *boiler efficiency* is often taken to mean the overall efficiency of the whole unit; that is, the percentage of the higher heating value of the fuel which will be in the steam. Efficiency is also designated by "evaporation," which is simply the pounds of steam produced per pound of fuel fired. This evaporation may be either the actual evaporation or the *equivalent evaporation* that would be obtained with the same heat, producing dry and saturated steam at 212° F. from feed water at 212° F.

The performance of a steam generating unit cannot be gaged by visual inspection of the equipment in operation. Yet an understanding of where

the heat units are going, what portion of them are usefully retained and what portion lost, and of those lost, how they are lost, and whether the losses are tending to increase, or whether they may be reduced, is highly desirable whenever there is an attempt to operate a steam generator with the highest standards of technique and economy. Furthermore, a breakdown of costs often requires definite numerical data as to the performance of the steam generating unit. Then, too, there is always the natural curiosity of operators as to the

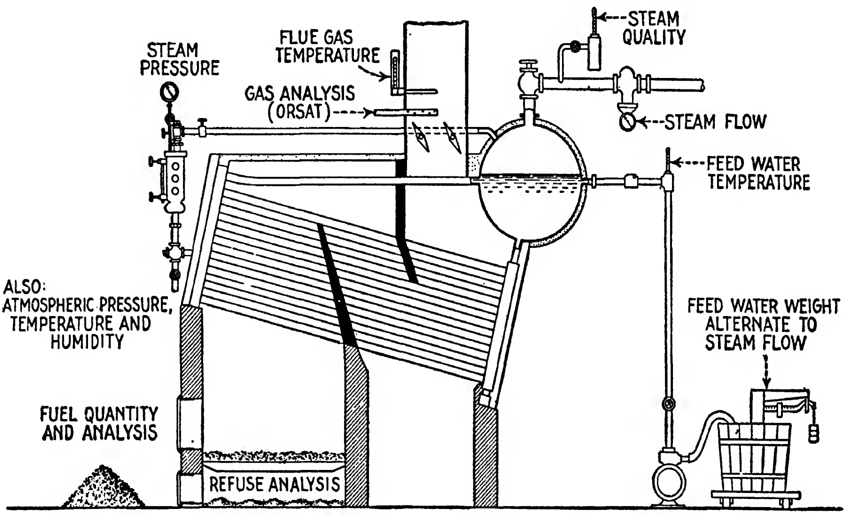


Fig. 13–19. Location of boiler test data.

results they are getting from their equipment, and whether these results are the best that can possibly be obtained. This information can be secured by a test on the unit, although a complete test is a job of no little magnitude. The test code is, in reality, a form for the arithmetical balance of heat entering a steam generating unit against the heat leaving. The heat in the coal or other fuel fired into the furnace must be accounted for either as useful heat or as one or the other of several different losses.

The standard test code for stationary steam generating units has the following items:

1. Heat absorbed by the water.
2. Loss due to moisture in coal.
3. Loss due to water formed by burning of hydrogen.
4. Loss due to hygroscopic moisture in air.
5. Loss due to heat carried away in dry chimney gas.
6. Loss due to incomplete combustion of carbon.
7. Loss due to unconsumed combustible in refuse.
8. Loss due to unconsumed hydrogen and hydrocarbons, radiation, and other losses.

The sum of these eight items should be equal to the higher heat value of the fuel. The test information that must be obtained, and the usual location of it in and about the steam generating unit, will be found in Figure 13–19. The test is made over an extended period of time to eliminate the effect of small irregularities of load, and no test of less than one hour's duration should be considered worth while. Tests are more frequently run for a period of 3 to 12 hrs. The accumulated data can then be used to calculate the magnitude of the losses. The computations involved are beyond the scope of this survey, but will be found adequately explained in any good book on heat engineering.

Example 1: A 3-hr. boiler test at 125 psi. gage pressure yielded the following data. Quality, 98%. Coal fired, 1620 lbs., heating value 12,650 B.t.u. per lb. Steam evaporated was measured by a flow meter, the load being controlled so as to be as constant as possible. Average flow meter reading, 5200 lbs. per hr. Average feed temperature, 185° F. What was the "boiler and furnace" efficiency?

$$\text{Water evaporated per pound of coal} = \frac{5200 \times 3}{1620} = 9.63 \text{ lb.}$$

$$\text{Enthalpy of the steam (tables)} = 321.6 + .98 \times 870.7 = 1174.9 \text{ B.t.u./lb.}$$
$$\text{Enthalpy of the feed water} = 152.9 \text{ B.t.u./lb.}$$

$$\text{Gain of enthalpy} = 1022.0 \text{ B.t.u./lb.}$$

$$\text{Thermal efficiency} = \frac{9.63 \times 1022.0}{12{,}650} = 77.9\%.$$

This is "boiler and furnace" efficiency.

Example 2: How much steam can a ton of Indiana bituminous coal (Table 3–1) produce from feed water at 120° F? Boiler pressure 95 psi. abs. Quality 99%. Efficiency 73%.

$$\text{Enthalpy gain of water} = 294.6 + .99 \times 891.7 - 87.9 = 1089.4 \text{ B.t.u. per lb.}$$

$$\text{Calculated heating value of coal} = 14{,}540 \times .6236 + 62{,}000\left(.0539 - \frac{.1550}{8}\right)$$
$$+ 4000 \times .0479 = 11{,}360 \text{ B.t.u. per lb.}$$

$$\text{Estimated evaporation} = \frac{11{,}360 \times 2000}{1089.4} \times 73\% = 15{,}270 \text{ lbs.}$$

PROBLEMS

1. Plot a t–h graph showing the process that occurs in a boiler which receives water at 120° F and delivers steam of 98% quality at 150 psi. pressure. Scales 1 in. = 100° F, 1 in. = 200 B.t.u. per lb.

2. Plot a T–s graph showing what occurs in a steam generator which receives water at 300° F and delivers steam at 450 psi. pressure and 600° F temperature. Scales 1 in. = 200° R, 1 in. = .3 B.t.u. per deg.

3. Using the data of Problem 2, determine what proportion of the heating is done in the superheater.

4. Find the K B.t.u. represented in the generation of 8500 lbs. steam per hr. from feed water at 210° F. Pressure is 250 psi., steam dry and saturated.

5. Steam is produced at 900 psi., 950° F, from feed water at 350° F. What is the K B.t.u. rating when generating 50,000 lbs. per hr.?

6. Estimate the coal required per hour for the installation of Problem 4, assuming coal No. II of Table 3–1, and an overall efficiency of 75%.

7. Estimate the coal required per hour for the installation of Problem 5, assuming coal No. III of Table 3–1, and an overall efficiency of 87%.

8. How much steam can be generated by a ton of coal if 72% of the heating value is transferred to the water? Heating value 12,400 B.t.u. per lb., steam pressure 150 psi., quality 99%, feed water 90° F.

9. At the outlet of a steam generator, the state of the steam is 300 psi. gage pressure, 500° F temperature. Feed-water temperature 150° F. Heating value of fuel (22° Baumé oil), 19,000 B.t.u. per lb. Efficiency of the steam generator 80%. How much steam can be produced per gallon of fuel?

10. Classify the boilers of Figures 13–3, 13–5, 13–7, and 13–10 on all of the bases mentioned in Section 13–2 that are found to be pertinent. Arrange answers in tabular form.

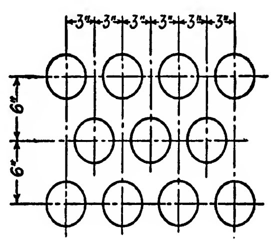
11. Same instructions as for Problem 10, but for Figures 13–4, 13–6, 13–8, and 13–9.

12. Compute the rated horsepower of the boiler of Figure 13–3. Allow one-half of the cylindrical shell as heating surface, but neglect any contribution from the tube sheets. $\frac{3}{16}$-in. thick tubes.

13. Draw a skeleton diagram (side elevation) of the steam generator of Figure 13–10. Emphasize the baffles. Put in arrows to show the path of the gases.

14. Assume that a boiler similar to that shown in Figure 13–9 has a tube bank six tubes deep by twelve tubes wide, consisting of 3-in. diameter tubes, each 16 ft. long. When operating at 150% rating, what is the developed boiler horsepower?

15. A boiler like Figure 2–4, with 4-in. diameter tubes, $\frac{3}{16}$-in. walls, will be built in a shell 48 in. in diameter. The tubes are spaced as shown here, and are 8 ft. high.



Lay out the tube sheet to scale 1 in. = 10 in. How many rated horsepower do these tubular heating surfaces represent?

16. What is the equivalent evaporation "from and at" 212° (see page 352) of a boiler which generates 1250 lbs. steam per hr., from feed water at 80° F? Steam at 100 psi., dry and saturated.

17. Prove that 34.5 lbs. of steam generated "from and at" 212° per hr. is the equivalent of 33,500 B.t.u. per hr. (rounded off).

18. A 1500 sq. ft. water-tube boiler generates 7200 lbs. steam hourly at 300 psi. dry and saturated from 210° water. Find (a) K B.t.u., (b) equivalent evaporation, (c) developed boiler horsepower, (d) per cent rating.

19. A boiler operates at 350 psi. pressure. Furnace temperature 1800° F, chimney temperature 500° F. Calculate the logarithmic mean temperature difference.

20. What is the logarithmic mean temperature difference for a superheater? Steam pressure 350 psi., superheated steam temperature 600° F. Flue gas before superheater 1900° F, after superheater 980° F.

21. What percentage of the heat added per pound of water in a steam generator was obtained in the superheater for each of the following cases? Feed-water temperature 250° F. (a) 100 psi., 100° superheat, (b) 400 psi., 650° F, (c) 900 psi., 950° F.

22. Given the following solid furnace wall section. Fire brick $4\frac{1}{2}$ in., red brick 8 in. Outside wall temperature 140° F, inside 1500° F. Conductivities from Table 4–1. Calculate the heat loss per square foot per hour.

23. An air-cooled furnace wall similar to *c*, Figure 13–14, has a 9-in. fire brick layer. Measurements show that 800 B.t.u. are received by the cooling air per square foot of air heating surface per hour. Estimate the conductance of this layer of brickwork, allowing U of air film, 2.0, of furnace film, ∞. Calculate the temperature drop through the fire brick itself, and from furnace to air.

24. A furnace wall such as *b*, Figure 14–13, is expected to have enough insulation to keep the outside casing at 130° F when inside wall is 1800° F. Conductivities of brick from Table 4–1, of casing ∞, of insulation .06 B.t.u. per sq. ft. per deg. per hr. per ft. Assume that the casing will discharge 120 B.t.u. per hr. to the atmosphere. How thick should the insulation be?

25. What furnace volume should be sufficient for burning 4 tons per hr. of a coal analyzing C 72, H 5, O 10, N 2, S 2, Ash 9? Rate of heat liberation 20,000 B.t.u. per cu. ft. per hr.

26. The volume of a furnace is 240 cu. ft. It contains grates having an area of 36 sq. ft. When coal is burned on these grates at the rate of 20 lbs. per sq. ft. per hr., what heat liberation rate exists? Heating value of coal 12,500 B.t.u. per lb.

27. The pipe joining pulverizer and burner (see Figure 13–2) should have an air velocity of 60 ft. per sec. to prevent flame back-travel in it. The pulverized coal can be transported upon 15% of the air. Air temperature 150° F. Theoretical air-fuel ratio 12:1. 20% excess air used. How big (dia.) should this pipe be, if it leads to a burner which is to fire 4000 lbs. per hr.?

28. Five tons per hour of Indiana bituminous coal are burned on a stoker with 60% excess air. Temperature of air entering stoker plenum chamber is 90° F. How many cubic feet per minute must be supplied? Neglect effect of plenum on density.

29. What type of furnace and firing equipment appears in Figures 13–5, 13–7, and 13–10?

30. What type of furnace and firing equipment appears in Figures 13–4, 13–8, and 13–12?

31. Results of a 5-hr. boiler test at approximately constant load were: Coal burned 2.75 tons, heating value 11,560 B.t.u. per lb. Feed water used 49,000 lbs., temperature 160° F. Steam 200 psi., dry and saturated. What was the average thermal efficiency?

32. Pennsylvania anthracite coal (data, Table 3–1) is burned with 80% excess air on hand-fired grates. Chimney temperature 550° F. Estimate the per cent of heating value that is carried off in sensible heat by the chimney gases. The specific heat = .25 B.t.u. per lb. per deg. F.

33. What is the "boiler and furnace" efficiency where a gallon of fuel oil (18,000 B.t.u. per lb. heating value) produces 110 lbs. of steam at 250 psi., dry and saturated? Feed water 200° F, specific gravity of fuel .88.

34. Using the data of Problem 33, calculate the equivalent evaporation "from and at" 212° F per gal. of fuel.

35. Assume that all the carbon, C, in a pound of coal appears either in X lb. CO_2 or Y lb. CO in the flue gases. X and Y are the fractional weights in a pound of dry flue gas. If X and Y are broken down and their carbon content C_x, and C_y established, then $C_x + C_y$ = carbon per lb. flue gas. The principle of "continuity of mass" implies that $C_x + C_y$ was continuously derived from C. Therefore

$$\frac{C}{C_x + C_y} = \frac{\text{lbs. dry flue gas}}{\text{pound of coal burned}}.$$

Use this principle to find the air-fuel ratio from flue gas analysis. The coal contained 72% carbon, 5% hydrogen, and 8% ash. A pound of flue gas analyzed 12% carbon dioxide by weight and .15% carbon monoxide. Note that dry flue gas + water vapor = air + coal − ash (conservation of mass).

36. Coke, which analyzed 85% carbon, was burned in a furnace producing a flue gas whose volumetric analysis included 10% CO_2. Assume that the mean molecular weight of the flue gas is 30.0. (a) Find the per cent of CO_2 present by weight. (b) Find the pounds of flue gas per pound coke burned. (Read problem 35.) (c) What is the air-fuel ratio?

37. The following coal is burned on grates with 60% excess air. Chimney temperature 650° F. What per cent of the heating value is lost as sensible heat up the chimney? Assume specific heat of the gas 0.25 B.t.u. per lb. per deg. C 70%, H_2 4%, O_2 15%, ash 11%.

38. Assume that Figure 13–19 represents the set-up of a boiler test using Indiana bituminous coal. The fuel fired during a six-hour test weighed 1875 lbs. A record was made of the feedwater pumped from the weighing tub. The cumulative total for the six hours was 15,675 lbs. The feedwater was uniformly 73° F. The thermometer on the steam calorimeter averaged 235° F. There being no calorimeter manometer, assume an open body throttling calorimeter. Pressure gage reading 125 psi. What do you find to be the operating thermal efficiency of this boiler?

39. How many square feet of water-tube boiler heating surface would be required to generate 10,000 lbs. steam per hour when operating at 150% rating? Feedwater 180° F. Steam dry and saturated at 100 psi. gage.

40. A horizontal return tubular boiler has 1000 sq. ft. of heating surface. It supplies steam for heating radiators. Steam is generated at approximately 5 psi. gage, 98% dry from water at 160° F. How many square feet of EDR (page 176) may this boiler be expected to serve when operated at 100% rating?

CHAPTER 14

Steam Power

14–1. Steam Prime Movers. Civilization uses mechanical energy to propel its vehicles on land and its ships at sea. The turning wheels and spinning motors of its industry, and the floods of energy poured continuously from the gigantic generators of its central power stations affirm again and again the pervading influence of mechanical energy in this modern world, for of such is its motivation. In deference to our present subject, one should know that most of this mechanical energy derives from heat energy via the path of steam power.

In Chapters 12 and 13 we have learned something of energy conversion through the use of a vaporous working medium, and of the means for creating heat energy in that medium. Now we are ready to examine ways and means for utilization of that heat—specifically, how and by what means it can be made into mechanical work. We begin, knowing that this is the function of a steam prime mover. Work is wrung from a heat-bearing vapor by an expansion. It is a simple matter to produce an adiabatic expansion, for steam, unlike internal combustion gases, is not so hot that cooling jackets are needed. Steam expansions can be carried out in heat-insulated equipment. Except for effects due to friction and turbulence, these expansions could be isentropic. In theoretical analyses they are frequently so handled. The reader should recall that the mechanical work appearing during such an expansion is simply the change of enthalpy of the working medium.

Steam prime movers may be devised using both Scheme 1 (steam engines) and Scheme 2 (steam turbines), as developed in Section 4–5. Engines are pressure machines; turbines flow machines. Engines are reciprocating; turbines purely rotary. Engines are bulky; turbines compact. Engines are fast starting, and readily arranged for reversing operation, while turbines must be slowly started and cannot be put into reverse action.

The field of application of steam power covers almost every usage except aeronautical, where the I.C. engine still reigns supreme. However, the enormous airplanes projected for the future may call for propulsive energy of such magnitude that steam turbine power will be the answer.

14–2. The Steam Engine. The steam engine is, with few exceptions, double acting, and at present is most frequently built and used in sizes of less than 500 hp. Larger power units are generally installed as turbines. The

steam engine is characterized by moderate or low speeds (100–500 rpm.), the use of atmospheric exhaust (not exclusively), high starting torque, and reversibility. The excellence of the steam turbine when a large amount of power is to be generated, especially where the exhaust is at a high vacuum, and the high efficiency of the Diesel engine as a prime mover, have greatly restricted the field in which the steam engine is economically a superior prime mover. However, where a boiler must be supplied anyway, as for the generation of heating steam, the steam engine is usually superior to the Diesel engine as a source of power, and where exhaust pressures are high (often the case in industry, where the exhaust is process steam), the steam engine offers advantages which are not seriously challenged by the other prime movers.

The principal parts of a steam engine are:

1. The frame or bedplate. In a multi-cylindered engine, this takes the form of a crankcase to which the cylinders are attached, but in a single-cylindered engine, the cylinder is often integral with the frame.

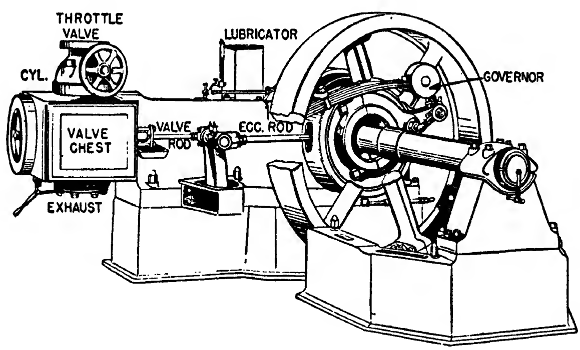


Fig. 14–1. Simple horizontal steam engine. (Courtesy Socony-Vacuum Oil Co.)

2. Cylinder, with valve chest.
3. Piston, piston rod, cross-head, connecting rod. This mechanical linkage receives a push from steam pressure at one end, and delivers it as a torque force on the crank.
4. Crankshaft, bearings, and flywheel. This part of the engine accomplishes the conversion of reciprocating to rotary motion, supports the shaft for power off-take, and steadies the speed.
5. Valve and valve gear. The device for admission and release of the steam to and from the cylinder, together with the means for actuating it from the crankshaft.
6. Governor. Stationary engines are automatically regulated for constant speed by means of a governor.
7. Lubrication. The piston and cylinder are lubricated by oil mixed with the steam. The bearings are lubricated with grease cups, oil rings,

wicks, etc. The cross-head is often lubricated with a sight feed oil cup. On certain engines the totally enclosed crankcase permits the use of a splash system of oiling.

Unless the steam pressure is especially high, steam engines will compare unfavorably in size with most I.C. engines of equivalent power. This is

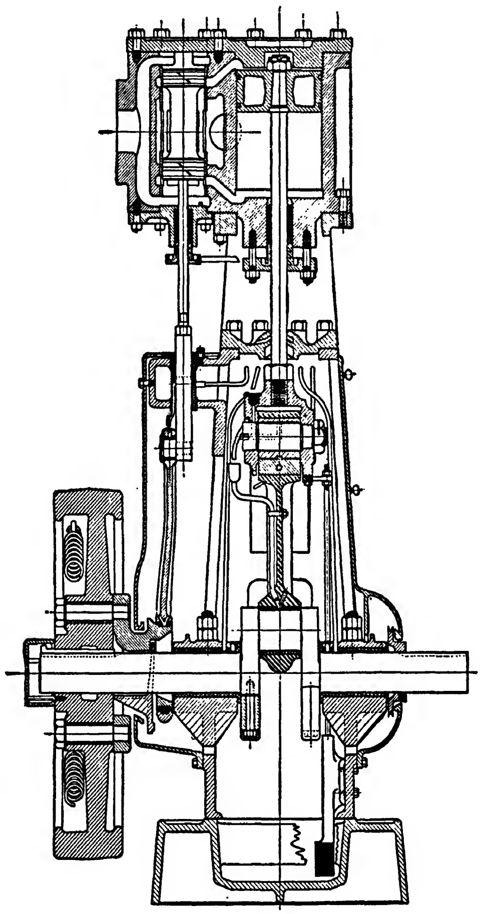


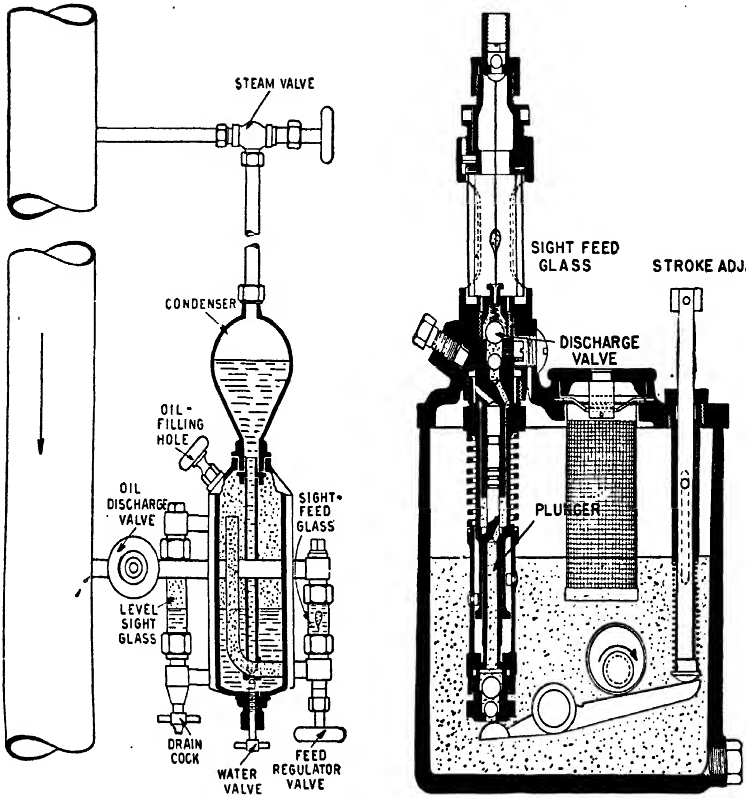
FIG. 14-2. Section through simple dual flow upright steam engine. (Courtesy Troy Engine & Machine Co.)

because the typical steam engine is a comparatively slow-speed prime mover. Power impulses are smoother, hence there is not the same trend to multi-cylindering. Typically, steam engines are single cylindered and are rarely found having more than three cylinders. An engine of conventional design, as shown in Figure 14-1, would have a horizontal axis and an enclosed crank-case. The flywheel is supported by a shaft which is carried by the crankcase and an outboard bearing. The steam valve is located in a *valve chest* on the

side of the cylinder and driven by an eccentric on the crankshaft. Live steam enters the valve chest through a throttle valve. The exhaust connection is underneath. The next figure shows a small vertical engine in section. Here the crankshaft is supported in bearings wholely provided by the frame. This section can be useful in furnishing ideas of the valve chest arrangement and of the relation between the cylinder and the remainder of the engine. Several of the subsequent engine illustrations will include only the cylinder and valve chest. It is the valve design that differentiates steam engines more than anything else. This engine has a piston valve. Other valve types are to be shown. Note especially the fluid seals for the piston and valve rods, the small clearance, the single port (per end), the eccentric drive to the valve, and lubrication. The latter feature is here obtained by a cross-head-driven plunger pump in the oil sump. The pump discharges into a reservoir (not shown) from which sight feed lines lead to the various bearings. Piston and valve lubrication is generally obtained either by a hydrostatic lubricator on the steam line, or by a mechanical force feed lubricator which forces oil into the steam at the valve chest.

A steam engine in which the total pressure drop of the steam between throttle and exhaust conditions is divided into two or more stages is said to be compounded when the expansion in each stage is performed in a separate cylinder. By common usage the engine in which steam is successively expanded in two cylinders is called a compound engine. If three stages are used, the engine is "triple expansion." There are three advantages of compounding an engine. (1) Any one cylinder does not have the big difference in temperature between incoming and outgoing steam that exists in a simple steam engine. Steam consumed in warming the cylinder and ports at the beginning of each stroke is thereby lessened. (2) Confining the highest pressure steam to the small high-pressure cylinder makes the large low-pressure cylinder easier to design. (3) The flow to the low-pressure cylinder may be split between two identical low-pressure cylinders, thus reducing the cylinder size.

A compound engine may be tandem-compounded or cross-compounded. Tandem compounding refers to the arrangement wherein the different cylinders are in line and their pistons are fastened to the same piston rod. One crank and one connecting rod are required in a tandem compound engine. The cross-compounded engine has cylinders set with axes parallel, each cylinder having its own piston rod and connecting rod, the latter bearing on separate cranks of the crankshaft. The compound engine was at one time built in very large sizes, and a quadruple expansion engine was no uncommon thing, but while the engine was efficient, it was bulky, slow in speed, and extremely costly, and has been rendered obsolete by the steam turbine. There are, however, many multiple expansion engines in service at present in excellent working condition, and they will probably continue to give good service for many



Hydrostatic type. Force-feed type.

FIG. 14-3. Methods of injecting lubricating oil into steam. (Courtesy Socony-Vacuum Oil Co.)

The hydrostatic lubricator is used to oil the high-pressure steam and so minimize friction on the slide valve and the piston. This type of lubricator is connected to the steam line supplying the engine and the body is filled with cylinder oil. The weight of the column of water forces the oil through the sight-feed glass and through the delivery pipe into the main steam line. A regulator valve adjusts the feed which can be observed, drop by drop, through the water in the sight glass. The level glass on the left of the reservoir indicates the quantity of oil still available.

Another method is known as force feed. This lubricator employs a single plunger pump for each feed line. Each unit is operated by an eccentric cam and lever. On the downward stroke, oil is drawn through the hollow plunger into the cylinder. On the upward stroke, the lower check valves prevent the return of this oil, which is then forced upward around the stationary central guide, and thence past the upper check valves into the sight-feed glass. By adjusting the screw shown at the right, the cam lever is raised toward or lowered from the eccentric, thus changing the stroke of the plunger. Here the oil rises through water of condensation in the sight-feed glass. For other lubrication the discharge would be arranged so that the oil dropped downward through a sight-feed glass, thence into a tube which will convey it to the desired point.

years. But new installations at present are made with either the simple steam engine or the steam turbine.*

14–3. Steam Engine Cycle. The admission and release of steam from a double-acting cylinder can be obtained with a single slide valve. The valve chest shown in Figure 14–4 contains a "D" slide valve which is reciprocated through a short stroke by the valve rod. A port leads from the chest to each end of the cylinder. The live steam inlet is just above the valve. The exhaust leaves from the region just below it. A displacement of the valve far enough

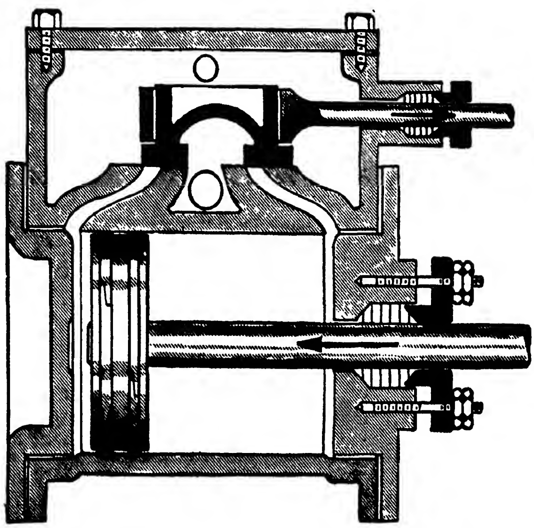


FIG. 14–4. Cylinder with simple D valve.

to the right will open the head-end port to the flow of live steam, at the same time bringing the crank end in communication with the exhaust region. Similarly, steam can be admitted to the crank end and exhausted from the head end with a left-hand displacement of the valve. It is also evident that, while the valve reciprocates at the same rate as the piston, *they do not move in unison.*

The cycle upon which the engine operates is briefly described as follows: Slightly before the piston reaches the dead center position corresponding to minimum cylinder volume, the valve connects the cylinder with the steam line so that as the piston starts on its outward travel, the full steam pressure is acting on it. The beginning of this action is known as the event of *admission.* The illustration shows this event on the head end. When some 20 to 30% of the stroke has been completed, the valve closes the port on the event known as *cut-off,* and during the remainder of the stroke, the steam is ex-

* Exceptions may sometimes be noted in locomotive practice. Also, some compound marine engines were installed when war-time needs could not be met entirely by turbine builders.

panded adiabatically to the accompaniment of decreasing pressure. Near the end of its stroke, the valve again opens the port, this time connecting the cylinder with the exhaust line. This event is known as *release.* The cylinder remains connected with the exhaust during the return stroke of the piston, and steam is expelled until approximately two-thirds of the stroke has been completed. The valve then closes the port, and the remaining steam is trapped in the cylinder and compressed for cushioning action. The beginning of this process is known as the event of *compression.* The four events just described govern the form of the steam engine cycle.

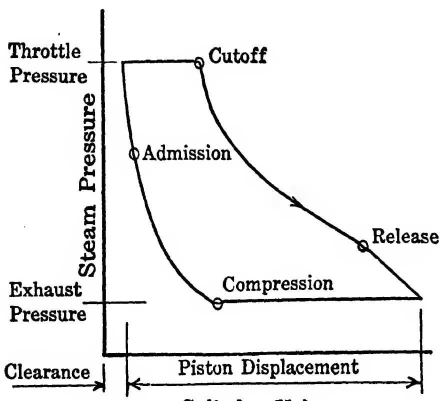


FIG. 14-5. Rankine engine cycle.

When these head-end events are spotted on the P–V plane and the processes which join them are plotted, the engine cycle appears as shown in Figure 14–5. An actual engine cycle, as taken by an indicator mechanism, would not appear with these events so sharply outlined. The corners of the cycle would be more or less rounded. The adiabatic expansion (and compression) of saturated steam has been found, by experience, to follow rather closely the equation $PV = C$. Contrary to previous use, $PV = C$ does *not* represent an isothermal process as in the case of polytropic gas expansions because saturated steam does not obey the gas laws. This is called the *Rankine Engine Cycle.* Do not confuse it with the Rankine Vapor Cycle (see Section 12–4).

While these processes are being performed in the head end, the crank end receives a similar cycle. In Figure 14–4 release has just occurred on the crank end. The valve does not jerk from one event to the next, but moves smoothly, being driven back and forth by an eccentric on the crankshaft. For example, after the event of admission, as shown, the valve will be moved to its extreme right position, then smoothly returned. When it registers in the same position it has in the figure, but with left motion, the event of cut-off occurs. If a small cardboard model of this figure is constructed, but having a loose valve, the complete head-end and crank-end action of the valve may be more readily visualized. The crankshaft timing diagram for a horizontal engine takes the form shown in Figure 14–6. Both head- and crank-end events are shown. A construction for relating crank angles of the events to the corresponding stroke percentages is also shown, using cut-off as the example. Notice that if the head- and crank-end angles are the same, the stroke percentages will

not be equal. As the *connecting rod:crank ratio* increases, the difference becomes smaller, but will not entirely disappear until the ratio is ∞.

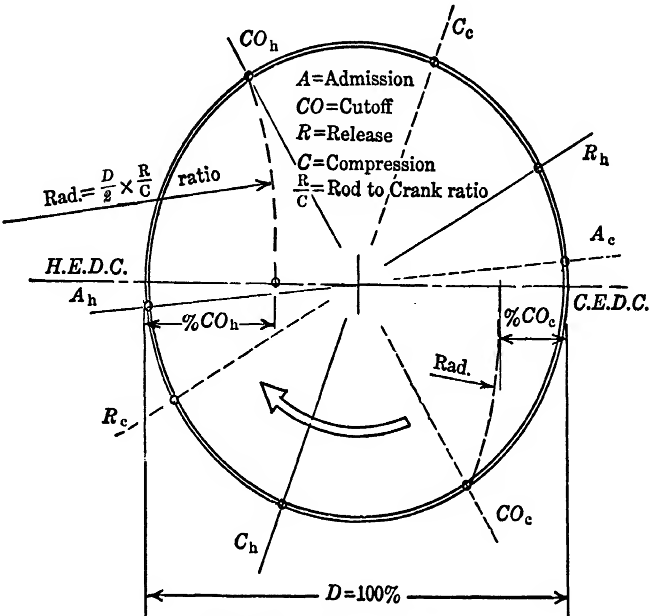


FIG. 14–6. Timing diagram.

An engine with cylinder ports as in Figure 14–4 is called a *dual-flow* engine because steam both enters and leaves by the same port. The elimination of exhaust steam flow over inlet ports is the major advance in steam engine de-

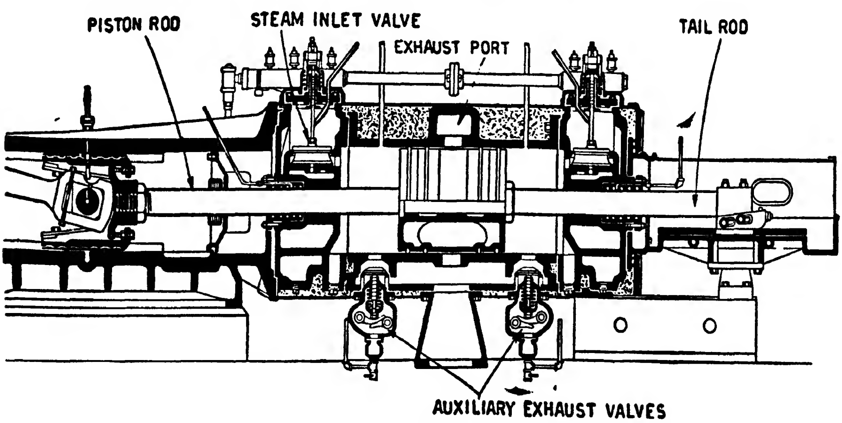


FIG. 14–7. Uniflow engine. (Courtesy Socony-Vacuum Oil Co.)

sign in recent years, because it eliminates initial condensation in ports and cylinder head. The term *uniflow* implies one-way flow of a fluid. It designates that improved type of steam engine in which the exhaust is arranged so

that the steam flows to exhaust ports located near the center of the cylinder, and does not reverse its direction of flow during exhaust, as is the case in the dual flow engine. The overcoming of initial condensation places the uniflow engine in a favorable position to compete with other types of prime movers. It is more expensive to construct, but so marked is its economy that it has come to be the only type of steam engine considered where a large efficient engine is wanted.

Except for the cylinder construction, the uniflow engine is similar to any other steam engine. The cross-section through the cylinder of a uniflow (sometimes written unaflow) engine shows how a ring of exhaust ports situated at the center of the cylinder will be uncovered by the piston as it nears the end of an expansion stroke. It also shows why the piston must be much longer in this type engine than in the dual flow.

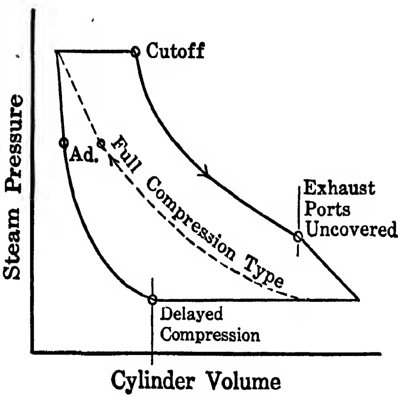


FIG. 14-8. Uniflow engine cycle.

On the return stroke the piston covers the ports fairly early in the stroke, and a considerable amount of work would be drawn from the flywheel in compressing the steam during the long back stroke were it not for the auxiliary exhaust valves which remain open a portion of the exhaust stroke, thus delaying the point of compression, and preventing the reabsorption of considerable work from the flywheel. This delayed compression does not produce higher thermal efficiencies than a full compression, but it increases the power which may be developed per cubic inch of piston displacement.

14-4. Valves and Gear. The steam engine valve gear exhibits more difference of design and detail than was found to exist in the I.C. engine field. This is because the valve gear not only produces the events of the cycle—but also varies them for governing purposes, and to obtain reversible action. The gear can be as simple as the eccentric and valve rod combination of Figure 14-2, or as complex as the locomotive valve gear of Figure 14-9. A simple slide valve engine has a valve action in which the valve motion is similar to, but always ahead of, piston motion. This will be obtained if the eccentric is given an "advance" of θ degrees on the crank, as shown in Figure 14-10. Since θ is required to be more than 90°, an angle $\theta - 90$ has been found more convenient to use. This is called the *angle of advance*, and is designated α. By being able to increase or decrease α, the events of the engine cycle may be altered in timing so that the work performed per cycle is varied. Decreasing

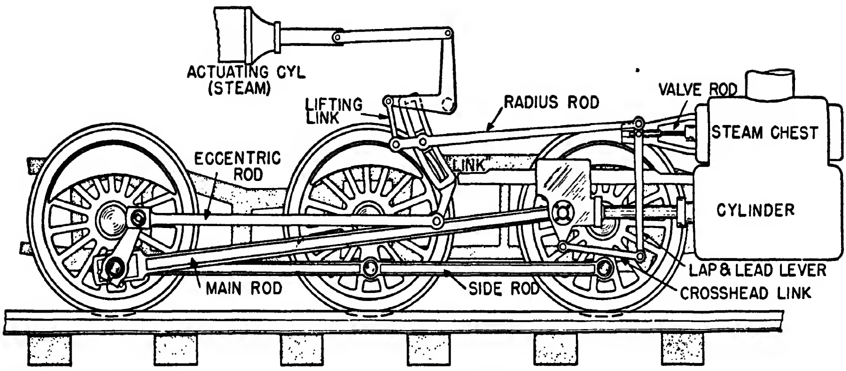


FIG. 14–9. Locomotive valve gear, Walschaert type.

α causes all events to occur later relative to piston motion, increases the area of the cycle on the P-V plane, and hence increases the work done.

If two eccentrics are provided, one ahead of the crank at $\theta°$, the other lagging by $\theta°$, then the engine could be operated either forward or backward,

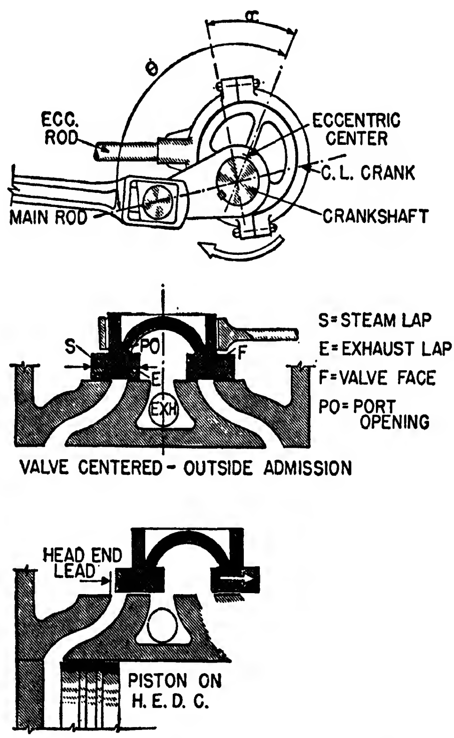


FIG. 14–10. Valve and gear nomenclature.

could the valve be driven at will from either of these eccentrics. This is the principle of the Stephenson gear, and is the simplest of all reversing gear.

The eccentric rods are pinned to either end of an oscillating "link." The link is held in space by a hanger bar pivot with which it may be raised or lowered. A horizontal motion-transmitting bar is pinned to a block that can slide in the link. Thus any motion the link has at the block swings the bar horizontally and thereby moves the valve. The mechanism is drawn in *neutral*, i.e., the block is midway on the link so the valve receives little or no motion. An auxiliary view shows what happens if the link is raised to bring the block near the forward eccentric rod. Valve motion is derived almost entirely from that eccentric, and normal forward power is produced. Conversely, if the link is

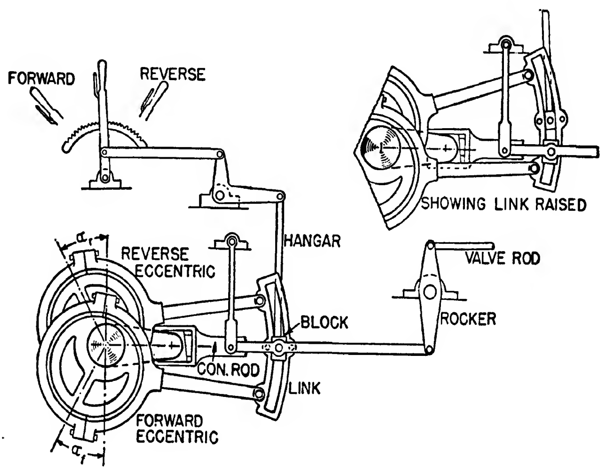


FIG. 14–11. Stephenson reversing valve gear (clockwise rotation, inside admission).

lowered to cause the block to register on the reverse eccentric, a valve action suitable for reverse power is had. Intermediate positions vary the amount of forward or reverse power.

Unlike the gasoline engine, in which the valves have become standardized on the poppet type, the steam engine is built with many different valve types. The original valve was known as the D slide valve, because in cross-section it resembled the capital letter D. This valve had definite disadvantages, to wit: the unbalanced steam pressure on it was difficult to handle except in low-pressure installations; its motion could be no faster than that of the piston, and consequently "wire drawing" of the steam existed during closure of the ports. Governing, by changing the point of cut-off, was possible only at the expense also of changing the other events of the cycle. The use of the D slide valve necessitated dual flow engines, that is, steam entering and leaving by the same port. The alternate heating and cooling of the ports causes the large thermal loss known as initial condensation. To overcome some or all of these disadvantages, a century of engine development has evolved the following improvements:

1. The piston valve—no unbalanced pressure.
2. The Corliss—no wire drawing. Partially balanced pressure, partially reduced condensation. Cut-off independently controlled. A slow-speed type with complicated gear. Not being currently built.
3. The poppet valve—balanced pressure, partially reduced condensation, and cut-off independently controlled.
4. Multi-port balanced valves—reduction of wire drawing. Balanced pressure secured by sealing off high pressure from back of valve.
5. Uniflow engine—elimination of initial condensation. Adaptable to any of the improved valves.
6. Double eccentric engines in which one eccentric throws a valve which controls admission and cut-off, the other a valve which produces the other two events. Only the first will be altered by the governor.

If a valve has high-pressure steam around the outside, and exhaust pressure on the inside, it is known as an *outside admission* type. Figure 14–4 is illustrative of the outside admission valve. Conversely, when the exhaust region surrounds the valve it is *inside admission*. Most piston valves have inside admission. Exhaust steam has much greater volume than inlet steam, and outside exhaust provides more space to accommodate it than can inside exhaust. See Figures 14–2, and 14–12 for examples of inside admission. In case of inside admission $\alpha = 90 - \theta$ and θ "lags" the crank.

14–5. Valve Motion Analysis. We have mentioned that angle of advance, valve shape, and port spacing all contribute to shaping the events of the cycle. In Figure 14–10 one can see what dimensions of the valve are involved. The *steam lap* is instrumental in the timing of admission and cut-off, the *exhaust lap* for release and compression. Several ingenious, geometrical constructions have been devised which show the interrelation of all the factors influencing valve motion and with which one can readily analyze the effect of a change of any factor. We shall show here the *Zeuner Diagram* because it is the best one * for general informational use.

To construct the Zeuner Diagram, first lay off two mutually perpendicular axes, shown by dash-dot lines in Figure 14–13. About O as center, and with radius OP equal to one-half the valve travel, draw the major circle. When the valve drive is direct, as in Figure 14–2, OP equals throw of the eccentric, but that would not necessarily be true were there a rocker arm as in 14–11. Lay off the angle of advance α counterclockwise from the vertical axis. Next the arcs of radii OS and OE should be drawn. OS = Steam lap. OE = Exhaust lap. PS will be the maximum port opening. Now erect perpendiculars to the angle of advance line at S and E and extend until they intersect the major circle at a and b, and c and d, respectively. Then considering OQ to be dead

* Other famous valve diagrams are the Reuleaux and the Bilgram diagrams.

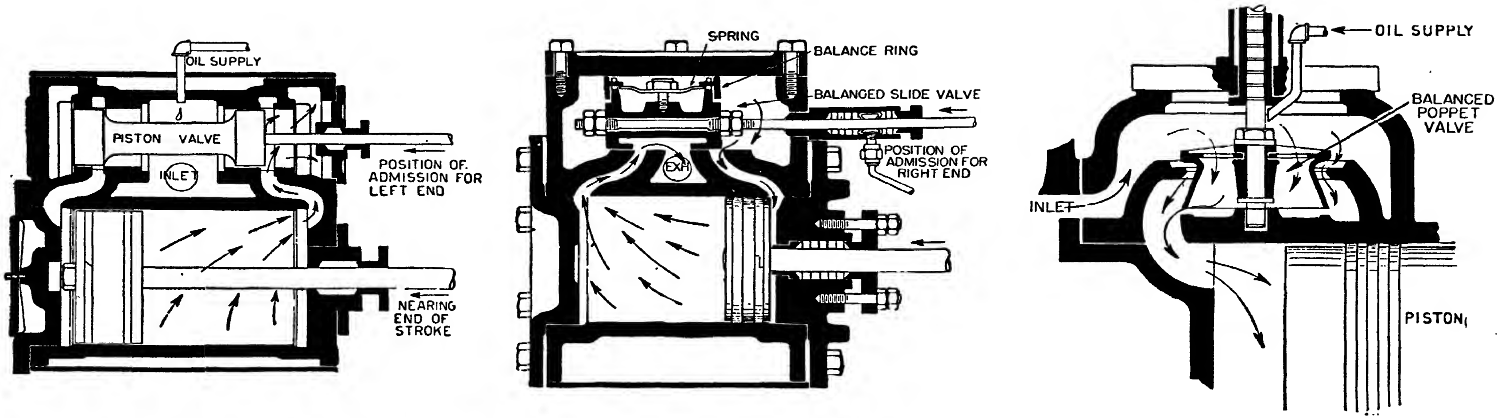


FIG. 14–12. Valve types. Although many other valve types can be found, these are common. All of these valves cure the D-slide valve defect of unbalanced steam pressure. The piston valve is commonly employed for locomotives. The balanced slide valve has a sealing ring moving with the valve and pressed, by means of a spring, against the valve chest cover. High-pressure steam leaking by the seal is relieved to exhaust. This construction dispenses with most of the pressure tending to force the valve against the seat. If a slug of water is trapped in the cylinder during compression the valve can lift from its seat to pass it. Piston valves are "inside admission." The balanced slide valve shown here is "outside admission." The reader can compare the figures and note the meaning of this term. On inside admission, steam lap is on the interior face of the valve; conversely, for outside admission. Inside admission is needed where inlet and exhaust steam vary enormously in volume. It requires the two exhaust paths be joined outside the ports. Poppet valves have no sliding against the seat. They are used where the steam is highly superheated. They are cam-operated, sometimes by rotating cams, sometimes by reciprocating cams. Poppet valves must be balanced (i.e., double beat) because, unlike I.C. engine valves, these must be opened against full operating pressure. Study of the construction will disclose that the lifting force exerted by the valve stem is a function of the difference of diameter of the two seats. (Courtesy Socony-Vacuum Oil Co.)

center position, the lines Oa, Ob, Oc, and Od are, respectively, the crank angles for admission, cut-off, release, and compression for a clockwise crank rotation. Percentages of stroke at these events could be found by use of the rod: crank ratio. This was done for cut-off in Figure 14–6, but is omitted here for the sake of simplicity. The radius of the small circle at the left end of the horizontal axis is the *steam lead*, i.e., the port opening for piston on dead center. Now if the two minor circles are drawn, each having a diameter of OP, additional information is yielded by the diagram. At *any* crank angle ϕ the valve travel

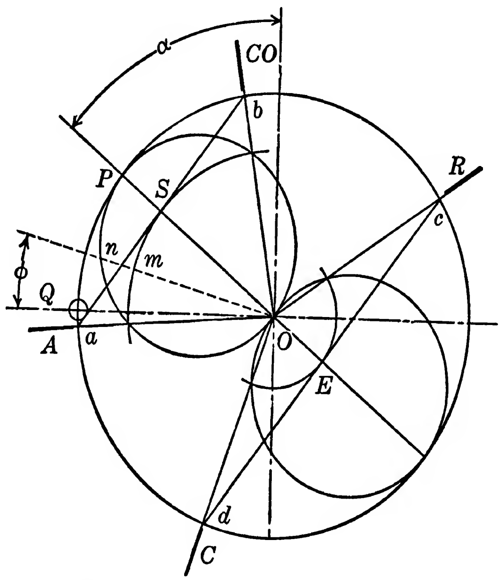


FIG. 14–13. Zeuner valve diagram.

from its central position is On; of which Om is steam lap, and mn is port opening. This diagram is quite useful in visualizing the effect, on the events of the cycle, of such actions as increasing α, decreasing steam lap, increasing lead, etc. It does not necessarily have to be drawn exactly in the order of this description. For example, the valve travel, cut-off angle, lead, and release might be stated for the purpose of finding α, steam and exhaust laps, and crank angles of the other events.

14–6. Control and Governing. As a usual thing, power needs are fluctuating. They may be grouped into cases where:

1. The speed is varied according to the need, and the power brought to whatever output will produce the speed. This is the case of the locomotive and the steamship. In the one, train speed is the variable in the other it is propeller shaft revolutions per minute.
2. The speed is held constant because the usage of power is best served at one particular speed. Thus the drives of generators, saw mills, etc., are best if the speed does not vary appreciably.

Case (1) is usually handled by manual control; case (2) by automatic governing. Manual control can be exercised by adjustments of the steam throttle or the valve gear, or both. The Stephenson and Walschaert gears, both of which have appeared on locomotives, are examples of manual control via the valve gear, adjustment being carried out at the link. There are two methods for automatic governing. In one, called cut-off governing, the per cent of the stroke during which the valve connects the cylinder with the boiler,

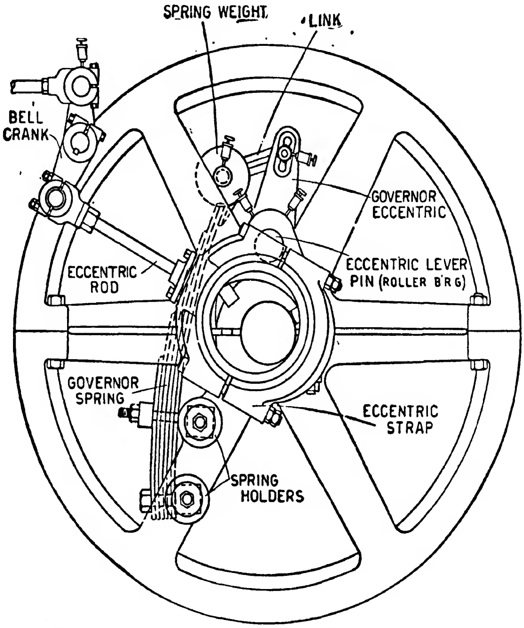


FIG. 14–14. Shaft-type governor. The rotation of the flywheel creates a centrifugal force that causes the spring weight to move outward against the tension of a spring. This movement is transmitted to the outer end of an eccentric lever, which is free to oscillate about a pin located on the flywheel. The inner end of the lever carries the steam-valve eccentric, which is free of the shaft. As the spring weight changes position, it alters the position of the eccentric, and thus changes the cut-off. This is more complicated but more efficient than a throttle-type governor. (Courtesy Socony-Vacuum Oil Co.)

is varied, and in this way different amounts of steam are admitted to the cylinder at one pressure. The mechanism to effect this type of control is incorporated in the valve drive. The other method, called throttling governing, consists of interposing an artificial resistance to create a pressure drop between the boiler and the engine, so that although the same volume is admitted on each stroke (the cut-off being constant), the weight of steam admitted will vary because of the variation in density created by throttling. The governor, in this case, operates on a throttle valve located at the steam inlet. The *a* and *b* governors of Figure 5–15 would be suitable for direct operation of a balanced throttle valve. Cut-off governors are usually built into the flywheel as in Figure 14–1. The diagram, Figure 14–14, shows an automatic cut-off governor. The rotation of the flywheel creates a centrifugal force that

causes the spring weight to move outward against the tension of a flat-leaf governor spring. This movement is transmitted by a link to the outer end of an eccentric lever, which is free to oscillate about a pin located on the flywheel. The inner end of the lever carries the steam-valve eccentric, which is free of the shaft. As the spring weight changes position, it alters the position of the eccentric, and thus changes the events. This is more complicated but more efficient than a throttle-type governor.

14–7. Steam Engine Power and Efficiency. Power developed by the steam engine is calculated in the same manner as for any other positive displacement engine using an expansible working medium. The reader is referred to Section 5–13, where the principles governing the computation of internal and external power are set forth. The typical steam engine might be said to operate on a two-cycle double-acting basis. The number of power strokes per minute per cylinder is therefore twice the number of revolutions per minute. Smoothness of action, resulting from overlapping power impulses in a two-cylinder steam engine, would be as good as in an eight-cylinder automobile I.C. engine.

Of course, initially the steam develops internal horsepower from an effective pressure acting on the piston face. The mean effective pressure can be varied (1) by throttling the inlet, keeping constant timing of the events, (2) by varying all the events, keeping throttle pressure constant, or (3) by varying cut-off and admission, but holding release, compression, and throttle pressure constant. Method (3) is better than (2), but requires a more complicated valve gear. Both (2) and (3) are thermodynamically superior to (1).

Example 1: The indicator card from a steam engine test is found to have an area of 4.75 sq. in. Its length is 2.9 in. The spring used in the indicator piston has *60* stamped on it. What was the mean effective pressure?

The average height of this card is 4.75/2.9 = 1.64 in. A 60 stamped on the spring means that the pressure scale of the indicator card will be 1 in. = 60 psi. So, the mean effective pressure is 1.64 × 60 = 98.3 psi.

Example 2: Analyze the following test data from a single-cylinder engine, 12 in. × 18 in. × 200 rpm. Piston rod diameter 2 in, head end m.e.p. 95 psi., crank end 110 psi. Dynamometer torque 4950 lb. ft.

Neglecting piston rod,

$$\text{IHP} = \frac{\dfrac{95 + 110}{2} \times \dfrac{18}{12} \times \dfrac{12^2\pi}{4} \times (2 \times 200)}{33{,}000} = 210.5 \text{ hp.}$$

Allowing for piston rod,

$$\text{Head end IHP} = \frac{95 \times \dfrac{18}{12} \times \dfrac{12^2\pi}{4} \times 200}{33{,}000} = 97.8 \text{ hp.}$$

$$\text{Crank end IHP} = \frac{110 \times \dfrac{18}{12} \times \dfrac{(12^2 - 2^2)\pi}{4} \times 200}{33{,}000} = 110.1 \text{ hp.}$$

$$\text{Engine IHP} = 97.8 + 110.1 = 207.9 \text{ hp.}$$

$$\text{External horsepower} = \frac{2\pi \times 4950 \times 200}{33{,}000} = 189.0 \text{ hp.}$$

$$\text{Friction horsepower} = 207.9 - 189.0 = 18.9 \text{ hp.}$$

$$\text{Mechanical efficiency} = \frac{189.0}{207.9} = 90.8\%.$$

A *conventional cycle* having no compression, clearance, or wire drawing, is sometimes assumed as a standard of comparison. This cycle is shown in Figure 14–15. This figure resembles Figures 12–5 and 12–6, but here *cd* represents a flow process in which steam is expelled from the cylinder, and not condensation as in the vapor cycles. The ratio of expansion $r = V_b/V_a$, and the mean effective pressure is

$$P_{\text{mep}} = P_1 \frac{(1 + \log_e r)}{r} - P_2,$$

in which P_1 = Throttle pressure.
P_2 = Exhaust pressure.

The *Diagram Factor* of a steam engine relates the mean effective pressures of actual and conventional cycles. This factor is defined as the ratio of the actual mean effective pressure to the theoretical effective pressure. The ratio is typical for any one class of engine. The accompanying table gives some values of diagram factor for steam engines. Knowing the steam conditions, and the type of engine, the probable mean effective pressure and horsepower realizable can be closely estimated by calculating the theoretical quantities and multiplying them by the diagram factor. In this light, diagram factor is a means of modifying theoretical calculations to bring them in line with actual experience.

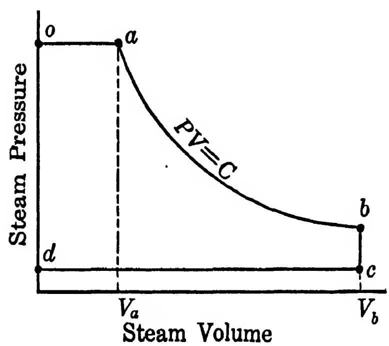


Fig. 14–15. Conventional Rankine engine cycle.

TABLE 14–1. DIAGRAM FACTORS

High-speed, simple automatic	0.70–0.85
Low-speed, releasing gear (Corliss)	0.80–0.90
Uniflows:	
Full compression, condensing	0.75–0.85
Full compression, non-condensing	0.70–0.80
Controlled compression, condensing	0.85–0.90
Controlled compression, non-condensing	0.80–0.85

A simple steam engine can convert into work a small part of the heat of the steam. This will range from 5% to 15%, depending on steam conditions. The remainder is distributed among these losses.

1. Heat remaining in exhaust steam. This is the largest loss, and is only moderately reducible. Use of a large P_1/P_2 ratio is helpful, but there is a limit to the increase of pressure ratio set by inability of the engine to handle excessively wet steam. Low quality steam (approximately 85% or less) will begin to shed moisture which upon accumulation in the cylinder in sufficient quantity will cause a "water knock" which can eventually grow to damaging severity if the condensation is not drained. Also, the exhaust usually contains more heat than necessary, because of the practice of releasing steam * prior to complete expansion.
2. Initial condensation loss. This refers to the condensation of some amount of the incoming steam on the walls of the port and cylinder head. The reason for this condensation is the cooling of the same surfaces by the outflow of the exhaust steam. Thus the ports have a cycle of heating and cooling which is in unison with the engine cycle. Unfortunately, the heat given up by the port during the cooling cycle is released to steam which is on its way out of the cylinder. Thus any initial condensation, though it may be re-evaporated, is an entire loss in so far as availability of the heat for work is concerned. This is no minor loss; in fact, it is one of the major reducible thermal losses of the steam engine. The uniflow engine, which represents the most significant advance in steam engine practice in recent years, derives its advantage chiefly from the elimination of this loss.
3. Wire drawing loss. This is a loss of work caused by throttling of the steam by slow-closing valves. It appears as the rounding off of the corners of indicator diagrams at the cycle events. Remedy: improved valve gear.
4. Mechanical and fluid friction loss. Steam engines have numerous sliding surfaces and bearings, few of which are ever of the antifriction type. The mechanical friction is therefore considerable.
5. Radiation and convection to the surroundings. This loss is quite small. The hot cylinders of these engines are insulated against heat transfer.

The rate at which a steam engine consumes steam per unit of power output is its *steam rate.* It is usually given in units of pounds of steam per horsepower-hour, or per kilowatt-hour in the case of a direct-connected turbogenerator unit. The steam rate varies with the load, being minimum at a load known as the most economical load. Steam rate is affected by point of cut-off, initial pressure, and degree of superheat; also, by exhaust pressure.

* See page 312.

Thermal efficiency is inversely proportional to steam rate. Steam rate, multiplied by load, is steam consumption.

The actual thermal efficiency, represented by performance, is:

$$\eta_t = \frac{2545}{w(h_1 - h_{f_2})},$$

in which w = Steam rate, lbs. per brake hp. hr.

h_1 = Enthalpy at throttle steam conditions.

h_{f_2} = Enthalpy of the liquid at exhaust pressure.

Where w is based on indicated horsepower, η_t is "indicated" thermal efficiency.

The numerator of the above equation represents the B.t.u. of energy in a horsepower hour, the denominator the heat that was supplied to produce the horsepower hour. The efficiency of an ideal engine is equivalent to that of an ideal vapor cycle.* The ratio of actual to ideal efficiency is a measure of the perfection of the engine for operating on the cycle for which it was designed. This is really the internal efficiency of the engine, and is one of its important engineering characteristics. It is commonly called "engine efficiency," but applies likewise to the steam turbine field where it bears the same title.† The Rankine engine or turbine will have an *engine efficiency* of

$$\eta_e = \frac{\eta_t}{\eta_R} = \frac{2545}{w(h_1 - h_2)},$$

in which h_2 is enthalpy of the steam after an *isentropic* expansion to the exhaust pressure.

Example 3: A steam engine which takes steam at 100 psi. dry and saturated and exhausts to atmospheric pressure is tested for steam rate while developing 25 hp. at the brake. It is found that 220 lbs. of steam were used during a 15-minute test. The thermal performance is to be analyzed.

$$\text{Steam rate} = \tfrac{220}{15} \times \tfrac{60}{25} = 35.2 \text{ lbs. per hp. hr.}$$

h_{f_2} is 180 B.t.u. per lb., and the Mollier Chart is consulted for h_1 and h_2.

$$h_1 = 1187; \quad h_2 = 1047.$$

$$\eta_t = \frac{2545}{35.2(1187 - 180)} = 7.2\%.$$

$$\eta_R = \frac{1187 - 1047}{1187 - 180} = 13.9\%.$$

$$\eta_e = \frac{2545}{35.2(1187 - 1047)} = 51.7\%.$$

Also

$$\eta_e = \frac{7.2}{13.9} = 51.7\%.$$

* Page 311.

† See also page 262.

14-8. The Steam Turbine. The reciprocating parts of engines limit their speed to a comparatively low value, but turbine energy is obtained from a number of small forces working at high velocity. Smaller dimensions and freedom from vibration give the turbine an advantage in first cost, space, and foundation requirements. Exceptions are noted in cases where power is small and operation is non-condensing.

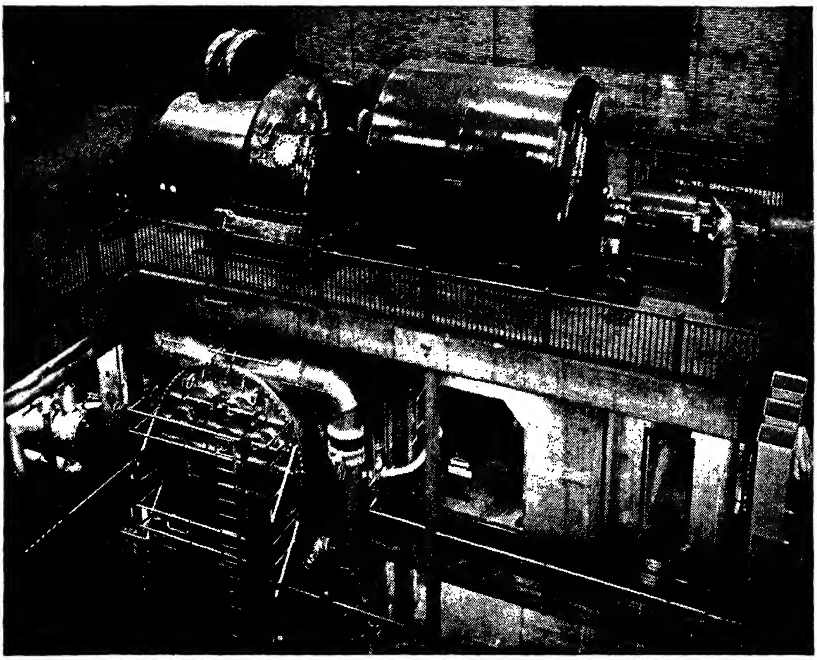


FIG. 14-16. Stationary steam turbo-generator unit. The turbine is the left-hand unit. The electric generator is direct-connected to it, while at the right an operator examines the exciter which is direct-connected to the alternating-current generator. The space beneath the turbine floor is occupied by auxiliaries. (Courtesy Allis Chalmers Mfg. Co.)

The action in a steam turbine is the transfer of energy from the heat form first to kinetic energy of a high velocity steam jet, then to the energy in the rotating shaft. Its principal parts are:

1. Nozzles to change heat energy to work energy, and to direct the course of steam onto blades.
2. Blades, which change the kinetic energy of the jet of steam into torque energy.
3. Rotating shaft, to which the blades are affixed and which delivers a rotating torque to the driven machinery.
4. A casing, which encloses the steam path and supports fixed parts.
5. Governor, bearings, lubrication, and other auxiliary devices.

It appears from the above that the nozzles and blades are the basic elements of the turbine. They accomplish the primary function of a heat → work transformation. The reader will recall that the energy of jets, as well as energy transfers by deflection of jets, was considered in a previous section of this book. He is asked to have in mind the contents of sections 4–8 and 5–4, as no repetition will be included here.

The moving parts of the turbine are few. Principally, there are two bearings, one at each end of the turbine, which support the rotor. These are rather

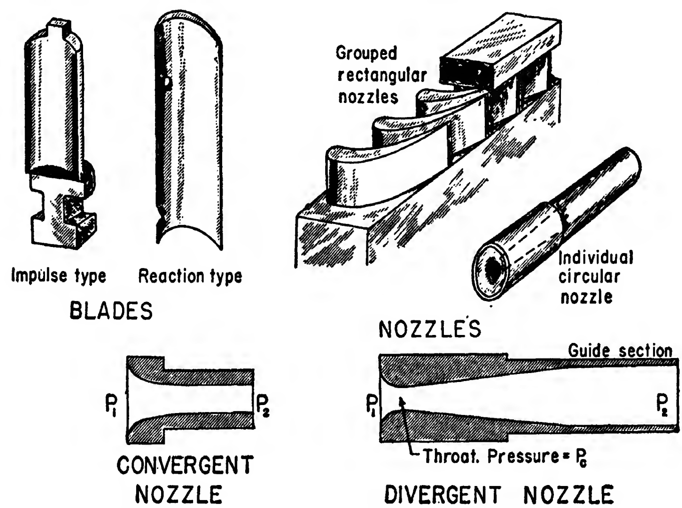


FIG. 14–17. Nozzles and blades. Nozzles must be (1) *convergent* if P_2 exceeds a certain critical pressure, P_c, or (2) *divergent* if P_c exceeds P_2. $P_c = 0.58P_1$ for saturated steam or $0.545P_1$ for superheated steam.

heavily loaded so that plain babbitted bearings are usual. Oil is pumped to these bearings and wasted from them to a sump, from which it is withdrawn, filtered, and cooled before being again supplied to the bearings. An auxiliary pump maintains oil pressure during the starting and stopping cycle of the main unit. Where the shaft of the turbine projects through the casing, means are provided for packing against leakage of steam outward at the high-pressure end, and infiltration of air at the low-pressure end. Sealing rings are used only on small turbines. This service is performed on large turbines by sealing glands which are built into the turbine, and which seal the shaft with steam or water. The outward leakage from the glands is minimized by labyrinths. Governing of steam turbines was mentioned on page 117.

There are two principles of action employed in turbines, known as the impulse and reaction principles. The impulse principle involves stationary nozzles and moving blades which absorb the mechanical energy from the steam as it flows over the blades. In the reaction turbine the nozzles are themselves attached to the shaft. In one case the motivating force is one of impulse of a

fluid stream against a blade; the other, one of a reaction force created by the acceleration of the stream in the moving nozzles.

The main field of application is the drive of constant speed apparatus, like generators, fans, and pumps. Yet they are also in marine service, where they are usually connected by reduction gearing to a slow-speed propeller shaft, and recently steam locomotives with turbine prime movers have started to appear on the railroads of this country. Nevertheless, these prime movers appear to best advantage where:

1. There is direct coupled drive.
2. The rotative speed is high (1200–3600 rpm.).
3. Operation is condensing.
4. High capacity is needed in a compact machine.

14–9. Impulse Turbine. The simple impulse turbine consists primarily of one or more nozzles, in parallel, and a bladed or bucketed wheel. The relation of these to each other is shown in Figure 14–18. The particles of steam

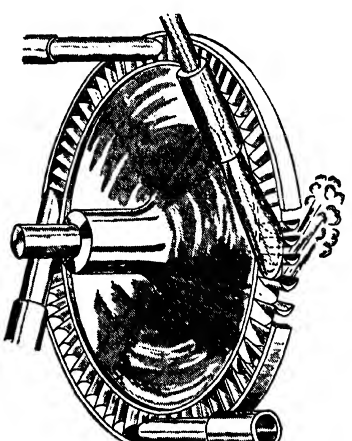


Fig. 14–18. Turbine wheel, blades, and nozzles.

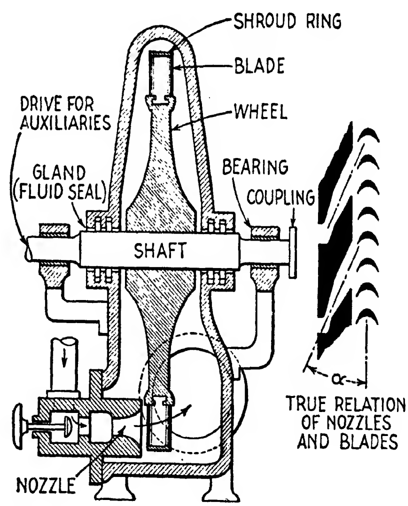


Fig. 14–19. Diagram of the simple impulse turbine.

receive an increase of kinetic energy at the expense of their heat energy as they shoot through the nozzles. Each nozzle aims its jet so that it will pass through the blades and be deflected thereby. We have already seen that this can exert a force on the blade, even though the blade is moving. A section through a turbine which would use such a nozzle-blade combination is furnished by Figure 14–19. As neither Figure 14–18 nor the section has shown the true angular position of the nozzles with respect to the blades, a development of the blading is supplied. Here it is seen that the nozzles are set at a

small angle α * to the direction of motion of the blades. The front of each blade is smoothly curved from edge to edge. A tangent to the blade face at the edge makes an angle β with the same reference line. Often these blades are symmetrical, then $\beta_1 = \beta_2$.

Now if the steam is to be most efficiently used it must move smoothly from the nozzle onto the blade without shock or turbulence, and be discharged with the minimum possible kinetic energy. The latter criterion will be met provided *all the jet's velocity in the tangential direction is consumed* by the deflection. To meet these conditions certain relationships are necessary. These are illustrated by the vector diagram of relative velocities, Figure 14–20.

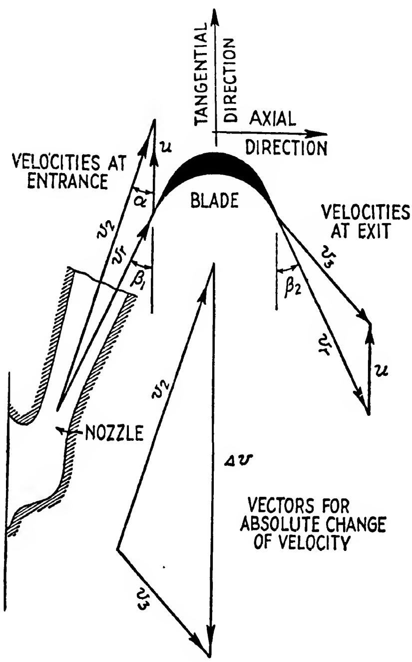


Fig. 14–20. Steam velocities for impulse blading.

A steam jet which leaves the nozzle at a velocity of v_2 will glide smoothly onto a blade which is moving at velocity u, provided the blade angle β_1 satisfies the vectorial subtraction $v_2 \rightarrow u = v_r$, as illustrated. The jet moving with a velocity relative-to-the-blade of v_r will be efficiently deflected by the blade to a final velocity of v_3. In finding v_3 it is here assumed that surface friction of the steam against the blade is negligible, and that there is no further energy-releasing expansion of the steam after leaving the nozzle. The relative velocity of the jet leaving the blade will have the same magnitude as at entering, but its direction will be governed by the exit angle β_2, to which it will conform. But since the exit edge is also moving forward at velocity u, the true exit velocity v_3 will be gotten by the vectorial addition $v_r + \!\!\rightarrow u$. The thrust imparted to the blades is $m\Delta v$, and the energy is $m\Delta vu$, in which m is mass flowing. As u is decreased, Δv increases, but the efficiency of energy transfer is maximum when v_3 is wholly axial in direction. This can be verified by considering kinetic energies of a pound of steam before and after deflection. Initially, the kinetic energy is $v_2^2/2g$; ultimately it is $v_3^2/2g$. Efficiency of transfer $= (v_2^2 - v_3^2)/v_2^2$, which is obviously maximum when v_3 is minimum.

* Although α was previously used to symbolize angle of advance, it is here employed for nozzle angle. This is in accordance with the common nomenclature of engine and turbine literature.

This will occur when v_3 is exactly axial, a condition which will be achieved if $u = \frac{1}{2}v_2 \cos \alpha$. Since α is a small angle, this practically amounts to the specification that u must be half of v_2 for maximum efficiency. Generally this will require the tangential speed to be too high and efficiency is sacrificed in favor of lower speeds.

Blade friction being negligible, the axial component of v_3 equals the axial component of v_2. When v_3 itself is axial, $v_3 = v_2 \sin \alpha$, therefore

$$\text{Maximum blade efficiency} = \frac{v_2^2 - v_2^2 \sin^2 \alpha}{v_2^2} = 1 - \sin^2 \alpha = \cos^2 \alpha.$$

The maximum possible efficiency of transfer of jet energy to blading is thus seen to depend only on the nozzle angle α. This should be as small as practical considerations permit. It is possible to use α's of 15° to 20°.

Example 1: Steam at 100 psi. expands through a nozzle to atmospheric pressure, then is deflected by symmetrical blades of *proper* shape which are mounted on a wheel rotating at 6000 rpm. Mean blade circle diameter is 30 in. $\alpha = 20°$. Find the work done per pound of steam, the blade efficiency, the thrust, and the flow required for 35 hp. Assume steam initially dry and saturated.

The Mollier Chart shows that an ideal nozzle expansion from 100 psi. to 14.7 psi. has $h_1 - h_2 = 1187 - 1047 = 140$ B.t.u. per lb. Substituting in the nozzle equation from Section 4–8, assuming v_1 negligible,

$$v_2 = 224\sqrt{140} = 2650 \text{ ft. per sec.}$$

$$\text{Tangential speed } u = \frac{30\pi}{12} \times \frac{6000}{60} = 786 \text{ ft. per sec.}$$

The vector diagrams of Figure 14–20 were laid out for the data of this problem. The graphical solution yielded $v_3 = 1290$ ft. per sec., $\Delta v = 3400$ ft. per sec. Thrust against the blade $= 1/32.2 \times 3400 = 105.5$ lbs. per lb. per sec. Since Δv is aligned with u, work $= 105.5 \times 786 = 82{,}800$ ft. lbs. per lb. Also, work may be calculated from the kinetic energy before and after deflection.

$$\text{K.E. work} = \frac{1}{2 \times 32.2}(2650^2 - 1290^2) = 82{,}800 \text{ ft. lbs. per lb.}$$

The above quantities are for a flow of one lb. of steam (or an m of 1/32.2).

$$\text{Flow necessary for 35 hp.} = \frac{35 \times 550}{82{,}800} = 0.2325 \text{ lb. per sec.}$$

$$\text{Efficiency of the blading in transferring energy} = \frac{2650^2 - 1290^2}{2650^2} = 75.8\%.$$

In the example just concluded, one point should be noticed. The operating conditions did not permit the best utilization of the available energy of the steam, as is attested by the considerable tangential velocity left in v_3. This, in spite of the uncommonly high wheel speed, u, employed, and the very

moderate steam expansion on the nozzle. Small turbines, especially those used to drive steam plant auxiliaries, and whose exhaust heat is salvaged by heating feed water with it, can justifiably be designed with low blade efficiency for the sake of simplicity. But where efficiency is important, the final velocity v_3 should be more nearly in the axial direction.

14–10. Multistaging. The foregoing remarks yield a clue to the need for "staging" steam action in a turbine. This is done by subdividing the total

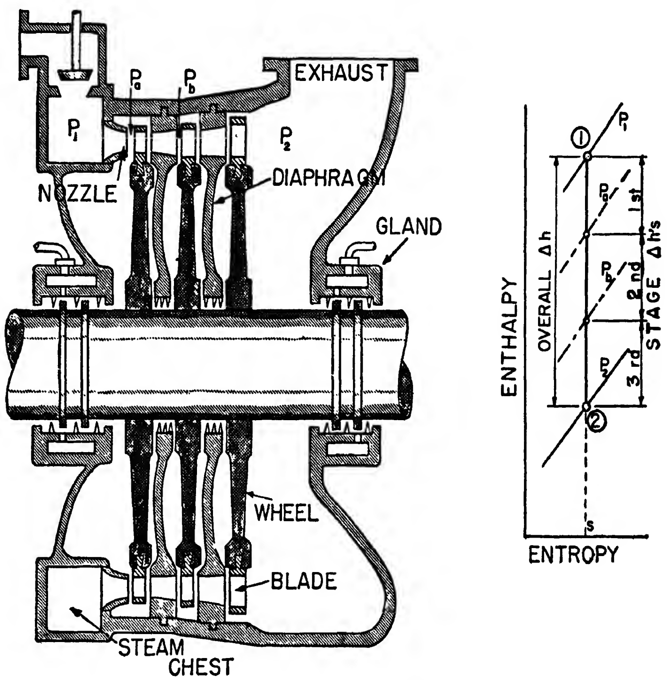


FIG. 14–21. Section through a three-stage impulse turbine.

pressure available for expansion, into stages so that the heat release per stage can be moderate. Each stage is virtually a turbine, like Figure 14–19, but all have these things in common. All wheels mount on the same shaft. All receive steam at the same rate because the same steam is passed from stage to stage. Furthermore, the outlet pressure of the first stage is the inlet pressure of the second, etc. The kinetic energy developed in a stage is absorbed before the steam is passed on, consequently the steam speeds are no more than those developed by the individual stage expansions.

Assume that steam is to be isentropically expanded in a turbine from an initial pressure of p_1, quality x_1, to a final pressure p_2. This process is the line 1–2 on the Mollier plane Figure 14–21. It yields a Δh which, it is also assumed, requires excessively high tangential wheel speeds for its efficient absorption in a single-stage turbine. Suppose we arbitrarily divide Δh into thirds and

propose that the processes occur in a three-stage impulse turbine, as diagrammed. The wheel speed will need to be only $1/\sqrt{3}$ * as large as in the single-stage case. Similarly, if five stages were used, the speed ratio could be reduced to $1/\sqrt{5}$. If enough stages are used, it is possible to have usable rotative speeds, like 1200–1800 rpm., and yet preserve high blading efficiency while using a considerable overall pressure drop.

Example 1: How many stages should be used in order to expand steam most efficiently with wheel speed of 3600 rpm. and diameter of 30 in.? $p_1 = 100$ psi. dry and saturated, $p_2 = 25$ in. Hg vacuum. $\alpha = 20°$.

These conditions are equivalent to

$$u = \frac{30\pi}{12} \times \frac{3600}{60} = 472 \text{ ft. per sec.}$$

For maximum efficiency, ideally,

$$u = \tfrac{1}{2}v_2 \cos \alpha. \quad \therefore v_2 = 1005 \text{ ft. per sec.}$$

But

$$v_2 = 224\sqrt{\Delta h}, \quad \Delta h = \left(\tfrac{1005}{224}\right)^2 = 20.2 \text{ B.t.u. per stage.}$$

Overall Δh from 100 psi. to 5 in. Hg abs. is obtained by use of the Mollier Chart. It is found to be 1187 − 959, or 228 B.t.u. Number of stages needed = 228/20.2 = 11+.

A turbine built for this pressure range would usually not have more than three or four stages, implying that builders usually compromise optimum steam conditions with cost of construction.

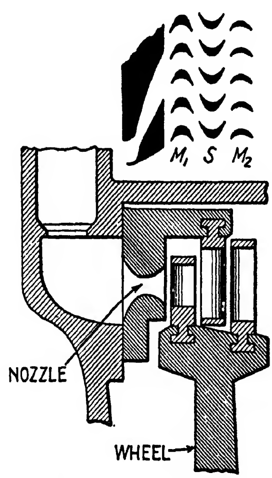


Fig. 14–22. Nozzle block and blades of Curtis staging. Moving blade rows M_1 and M_2 have the same speed. Steam speed is greatest over M_1, least over M_2.

The internal parts of a three-stage turbine are shown in Figure 14–21. Steam, admitted from a steam chest, passes directly into the first-stage nozzles, which expand it from p_1 to p_a. The resulting kinetic energy is absorbed by the first-stage wheel, then the steam enters the second-stage nozzles where the expansion is p_a to p_b, and so on until the exhaust is reached. The stages are kept separate by *diaphragms,* which are circular walls arranged to block off the interior of the casing into separate pressure regions. The nozzles which communicate from one region to another are set into these diaphragms. Fluid seals are needed at the intersection of the shaft and casing. The clearance between the diaphragm and shaft must be small so as to restrict steam leakage from one region to the other This type of multi-staging is called *Rateau staging.*

* Because $u \sim v_2$ and $v_2 \sim \sqrt{\Delta h}$.

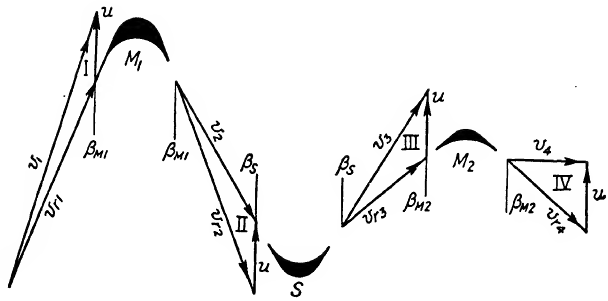


FIG. 14–23. Steam velocities in Curtis staging. The nozzle is assumed to have produced a velocity of v_1, while the moving blades M_1 and M_2 have a tangential velocity u. Blade rows are separated in order to accommodate the vector diagrams. Triangles I and II represent respectively velocities at entrance and exit of M_1; III and IV similarly for M_2. Velocities subscripted r are relative to either M_1 or M_2. Note the effect of steam velocities on blade shapes; also, as shown in Figure 14–22, on blade height.

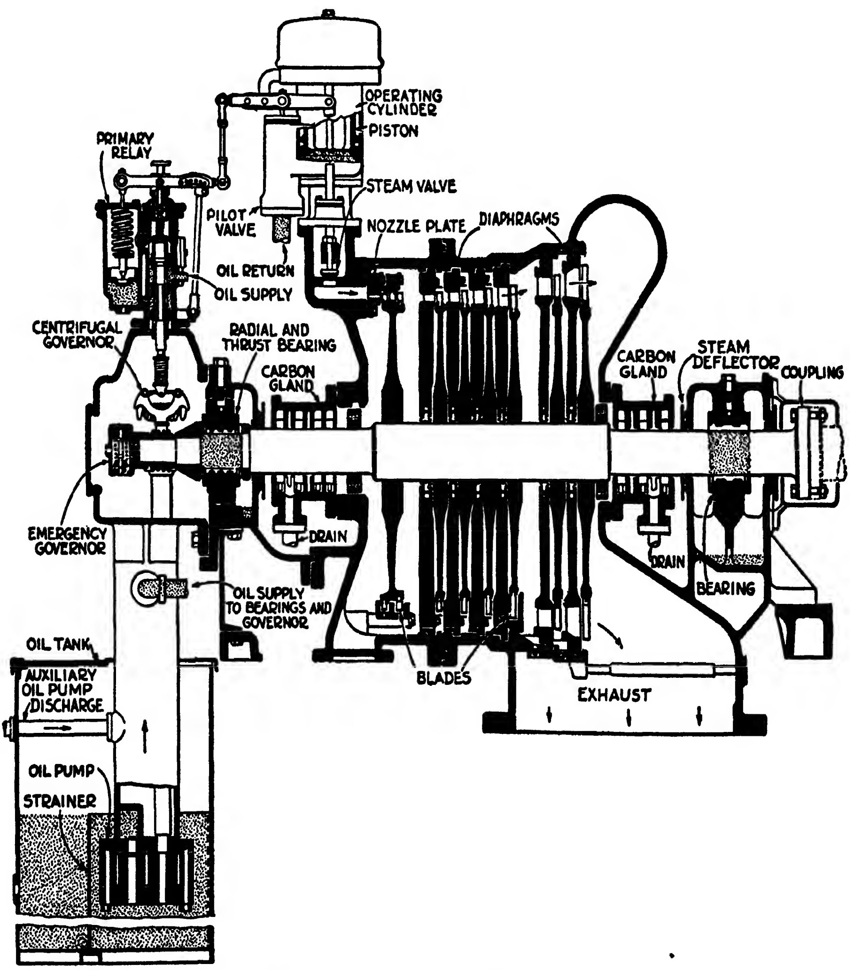


FIG. 14–24. Seven-stage impulse turbine (first-stage velocity compounded).

Another scheme for reducing tangential wheel speed is to absorb the kinetic energy generated by the nozzle on more than one group of moving blades, consuming velocity progressively. Such is called *Curtiss staging*, and is illustrated in Figure 14–22. Steam undergoes considerable expansion in the nozzle, and high steam speeds are generated. After deflection by the first row of blades the steam contains a large residue of kinetic energy which must be absorbed in the second row of moving blades. As both wheels are on the

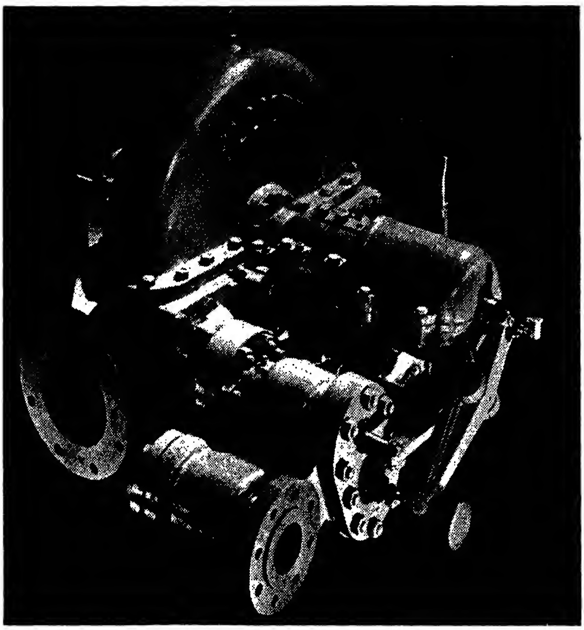


FIG. 14–25. Single-stage impulse turbine. Steam passes a strainer, governor valve, and overspeed trip valve before reaching the nozzles. (Courtesy Elliott Co.)

same shaft, both u's are in the same direction. An intermediate set of fixed blades must be installed to redirect the steam emerging from the first wheel, so that it will approach the second wheel in the proper direction. A two-velocity Curtis stage requires u to be only $\frac{1}{4}v_2 \cos \alpha$, a three-velocity stage $\frac{1}{6}v_2 \cos \alpha$, etc., for optimum steam flow conditions. It is obvious that a velocity stage may be inserted in any pressure stage. If instead of a simple single-stage turbine of maximum efficiency, we have three pressure stages, each containing a three-velocity stage, the multi-stage turbine speed would only have to be $\frac{1}{3} \times 1/\sqrt{3}$ that of the single-stage machine.

Figure 14–25 shows a small auxiliary type turbine suitable for direct connection to very high-speed equipment, or for connection to conventional equipment such as fans and pumps by means of a gear box. The governor is on the end of the shaft and operates the steam valve directly by means of a

rocker arm. In addition to the governor valve, there is an emergency valve which is tripped closed in the event of overspeeding.

The steam connection in the foreground contains a strainer. As all the connections are to the lower half of the casing, the top half is readily lifted,

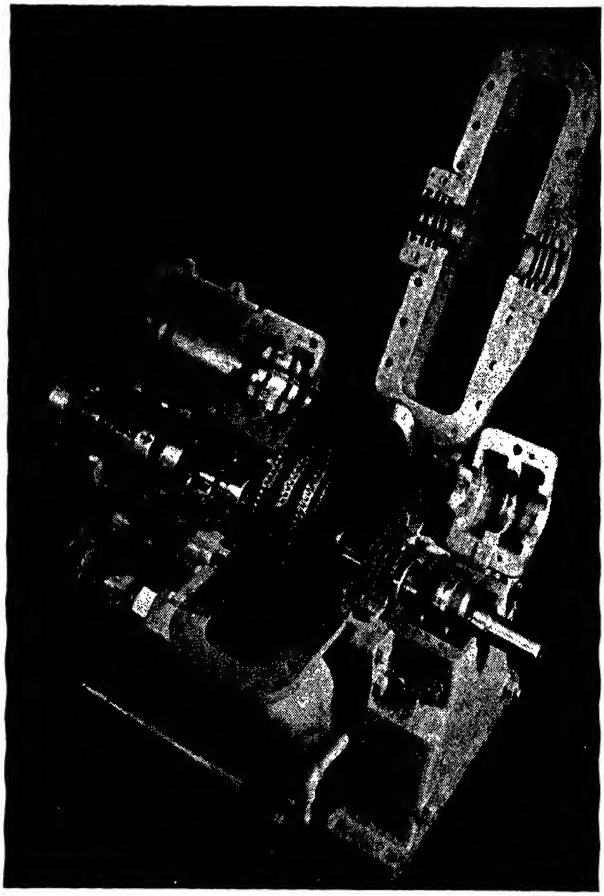


FIG. 14–26. Turbine of Figure 14–25 with top casings removed. Note the three wheels of the three-velocity Curtis-type blading, governor at left, nozzles and reversing blades in lower part of the casing. (Courtesy Elliott Co.)

revealing, in Figure 14–26, three wheels. Also visible are the governor weights, governor spring, bearings, and shaft packing glands. The nozzle block and attached stationary blades shown in Figure 14–27 indicate that this is a three-velocity Curtis stage.

Another type of velocity staging is known as *"re-entry" staging.* Instead of redirecting the steam onto another wheel, the intermediate blades (which now become curved conduits) direct it back onto the same wheel. Sometimes the steam is passed back and forth through the blades of one wheel several

times. This accomplishes velocity staging in a simple turbine, but suffers from low efficiency, for obviously the blade angles and heights are correct for but one of the stages.

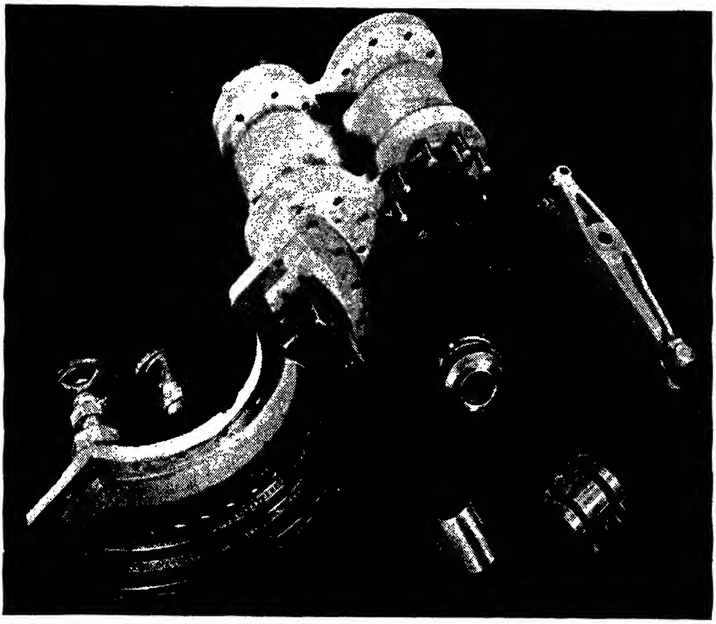


FIG. 14-27. Reversing blades and nozzle block for the turbine shown in Figures 14-25 and 14-26. (Courtesy Elliott Co.)

14-11. The Reaction Turbine. One of the earliest recorded steam-driven devices was a crude reaction turbine—the eolipyle, described by Heron of Alexandria. Crude as it was, the action is so obvious that we use it to illustrate the reaction principle (Figure 14-28). Steam generated under pressure below the sphere was introduced into it through a hollow axle. From thence it blew out to the atmosphere through nozzles fastened in such a manner that the reaction to the jet's impulse would whirl the sphere about its axle. So, in a modern turbine, moving nozzles throw out steam and receive a reaction which is passed on to the shaft or drum to which they are affixed. But here the similarity ends and the reaction steam turbine bears scant physical resemblance to Heron's eolipyle. The moving nozzles, for example, seem to be but rows of blading. The steam is *thrown* to them from stationary nozzles. Staging is practiced. After being thrown from a moving nozzle, the steam is caught by the stationary nozzles of the next stage and delivered to its moving nozzles,

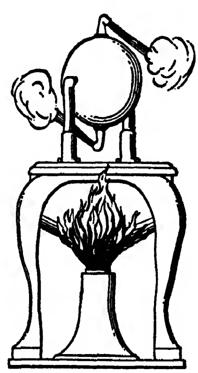


FIG. 14-28. Heron's eolipyle.

this going on sometimes through as many as twenty to forty stages. When the stages are numerous the stage Δh is small. In these turbines Δh is so small (10 B.t.u. approximately, half in the fixed, half in the moving nozzles) that the nozzles are always convergent. Now, as Figure 14–29a shows, spaced nonsymmetrical blades of the right profile can become a row of convergent nozzles. Having this in mind, we are ready to examine the typical reaction turbine blading arrangement shown in Figure 14–29b. There are no wheels, no diaphragms. The rotating element is a drum on the periphery of which are

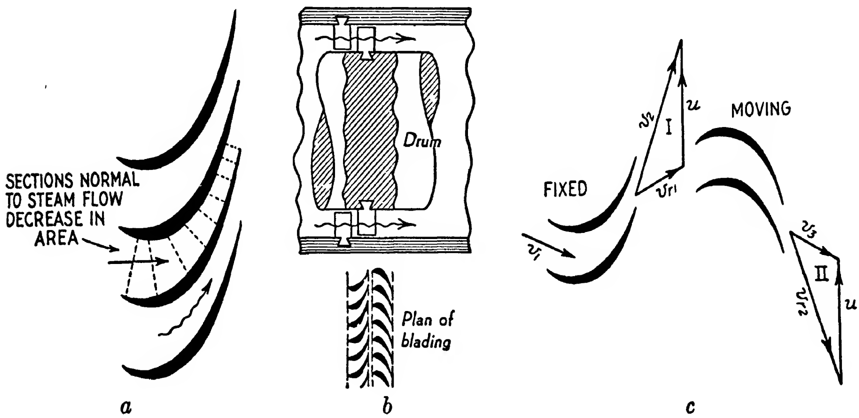


FIG. 14–29. Principle of the reaction turbine.

a. Showing how asymmetric blades become a series of converging nozzles.
b. One stage of a reaction turbine.
c. Steam velocities for reaction blading. Triangle I—Velocities at entrance to moving blades. Triangle II—Velocities at exit from moving blades.

mounted the moving blades (nozzles). These alternate with fixed blades attached inside the casing. Steam is admitted to the first group, which is a row of fixed nozzles, and is expanded some as it passes each row of blades. Consider the action in some typical intermediate stage consisting of a row of fixed blades followed by a row of moving blades. Passing through the fixed blading, the steam is expanded and given the speed v_2, which, in conjunction with the moving blade speed u, is sufficient to carry it smoothly into the entrance of those nozzles with a velocity which we here designate v_{r_1} because it is the velocity of approach to a nozzle. The expansion occurring while passing through the nozzle imparts an exit v_{r_2} such that $v_{r_2}^2 - v_{r_1}^2 = 224^2\Delta h_m$. Steam speeds are low in reaction turbines, and v_{r_1} cannot be neglected as in impulse theory. The absolute leaving velocity is v_3. Vectorally, $v_3 = v_{r_2} \mathbin{+\!\!\rightarrow} u$. Without attempting any specific demonstration or proof, the following statements conclude our survey of the basic features of steam action in this type of blading, which is called *Parsons staging*.

1. A proper combination of α, β, u, and Δh can make v_3 of the same magnitude and direction as v_1, thus

2. An identical steam action can occur in the following stage provided α, β, u, and Δh are the same Only pressure and blade height need vary from stage to stage, the pressure decreasing as expansion is continued, and the blade height increasing to accommodate the steam of greater specific volume.
3. At the design condition the discharge jet from a stage is taken in smoothly and without loss by the fixed nozzle of the following stage. A 100% blade efficiency is ideally possible (compared to $\cos^2 \alpha$ for the impulse type).
4. Since there is a pressure difference across the moving rows, they receive an axial thrust directed toward the low-pressure end.
5. The work delivered to the moving blades is no longer $(m/2)(v_2{}^2 - v_3{}^2)$, but $(m/2)(v_2{}^2 - v_3{}^2) + J\Delta h_m$. The thrust remains $m\Delta v$, as in the impulse type.

As steam expands through a multi-stage turbine it increases in volume after each stage, making it necessary either that the diameter of the circle in

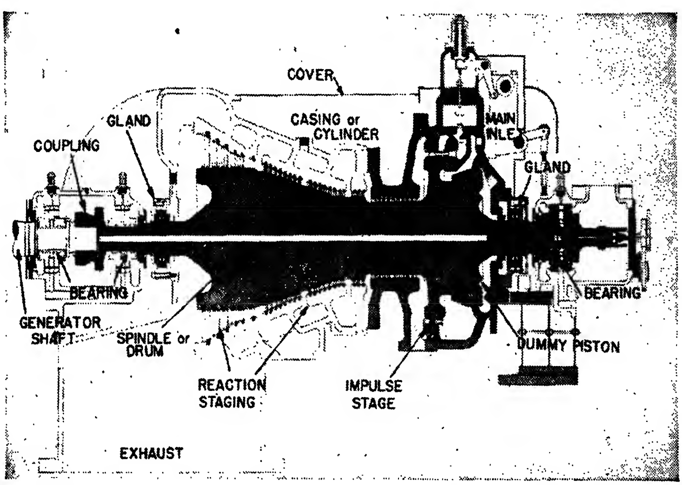


FIG. 14–30. Section of an impulse reaction turbine. When reaction staging is employed it is frequently preceded by an impulse stage of the Curtis type. The latter can take the place of several reaction stages in expanding steam, thus reducing the overall length. (Courtesy Allis Chalmers Mfg. Co.)

which the blades travel be increased, or that the height of the blades be increased to provide sufficient area for the increased volume of flow. Usually both of these expedients are adopted, so that the turbine exhibits, roughly, a somewhat conical shape, being smallest at the high-pressure end, and largest at the exhaust.

Reaction turbines have to be full admission, i.e., steam admitted to the first row of blades around the full circumference. Impulse turbines can be partial admission, and this can also help to pass the large steam volumes because

as lower pressure stages are encountered, the nozzles may extend around more of the periphery of the diaphragms.

Essential to the operation of a turbine are a number of auxiliary devices. The impulse turbine rotor receives a moderate end thrust, due to fluid friction on the blades. The reaction turbine has a large end thrust, arising from

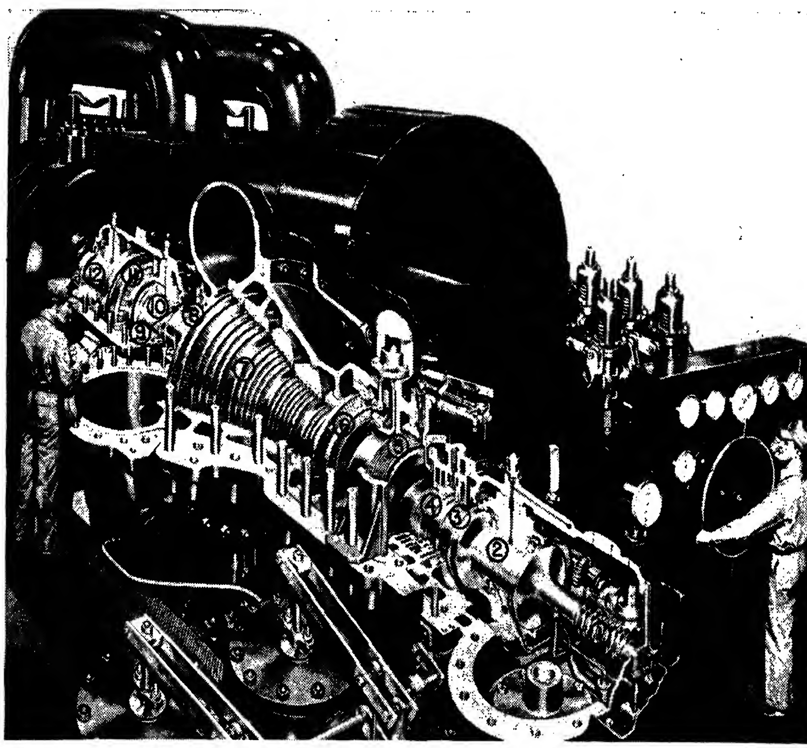


FIG. 14–31. Cutaway view of an impulse-reaction steam turbine. The blading arrangement is similar to that in Figure 14–30 except that here the turbine is tandem-compounded. Beginning at the right-hand end of the shaft, note the following in order: 1. Drive for auxiliaries. 2. Combined journal and thrust bearing. 3. Oil baffle. 4. High-pressure gland. 5. Dummy piston with labyrinth seal. 6. Impulse stage. 7. Several reaction stages. 8. Low-pressure gland. 9. Oil baffle. 10. Journal. 11. Coupling. 12. Journal of the low-pressure turbine. (Courtesy Allis Chalmers Mfg. Co. and Socony-Vacuum Oil Co.)

the drop of steam pressure, across the moving blades, acting on the blading annulus. While the end thrust of an impulse turbine can be accommodated by special thrust bearings, the large forces set up in a reaction turbine necessitate the use of a special balancing device known as a dummy piston. The dummy piston is a circular plate mounted concentric with the axis of the turbine, and having, on one side, high, and on the other, low steam pressure. The pressures are so chosen that the direction of the resultant force on the plate is counter to the end thrust on the blades. A separate piston is used to balance each different diameter of the drum, so that although steam pressure

and end thrusts may vary with different loads, the equalizing pressures also similarly vary. A thrust bearing is provided to absorb the small amount of unbalance which may still exist.

14-12. Compound Turbine. Some large turbines have two casings—high- and low-pressure. The steam is partially expanded in the high-pressure casing stages, then delivered to the low-pressure casing, where further expansion is carried out. The rotor arrangements may be either tandem- or cross-compound. Two generators must be supplied to cross-compounded turbines.

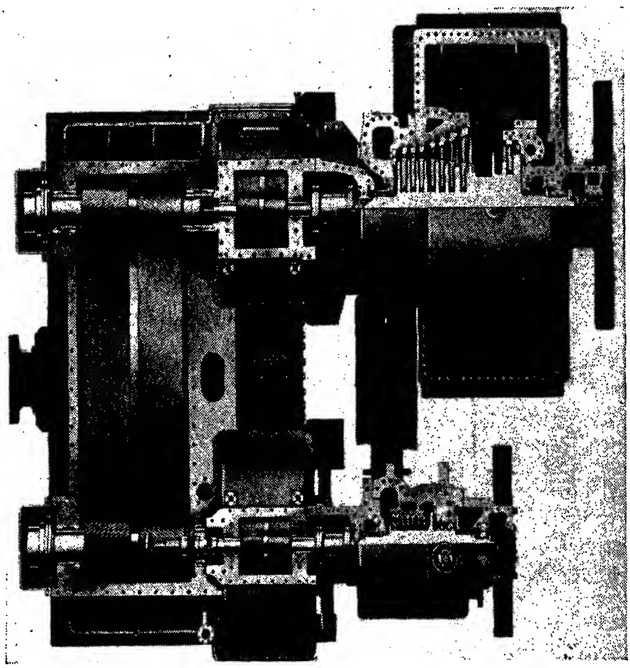


FIG. 14-32. Plan view of a cross-compound impulse turbine with speed reduction gears. Designed as a ship propulsion unit. (See also Figure 12-14.) (Courtesy General Electric Co.)

The principal advantages of compounding a turbine are the reduction in physical size of any one casing, the confinement of the highest pressures to the smaller casing, which may be built of steel, and the possibility of divided flow in the low-pressure casing for the purpose of equalizing end thrusts. Turbines for ship propulsion are compounded for still another reason—curtailment of overall length to suit the available space. The plan view of a cross-compound main propulsion turbine is shown by Figure 14-32. This is an impulse type consisting of Rateau stages. The two shafts bear the first stage pinion gears of a two-stage helical reduction gear set. The bull gear shaft is direct-connected to the propeller shaft. This is a "reversing" turbine, but only because a special reversed-blading group of stages is installed on the

low-pressure turbine shaft. For reversing, the main throttle is closed and the reversing section then energized.

14–13. Turbine Capacity and Efficiency. The power developed by a turbine depends on the quantity of steam passing through it and the kind of working expansion the internal design of the turbine gives to the steam. As the blades of steam turbines, once installed, are rigidly fixed in their sockets, any variation of the steam flow from that on which the turbine design was based creates turbulence which lowers performance. Yet the common method of decreasing power below design conditions is by throttling the inlet, an action which changes steam speeds, volumes, and relative jet angles throughout the turbine. More-than-normal power obtained by diluting partially expanded steam with some by-passed directly from the throttle also destroys streamline flow. Although high blade thrusts are obtained, the efficiency of energy transfer is diminished. A turbine, therefore, will have an operating point of maximum efficiency when inlet steam is unthrottled, and steam conditions internally correspond to the predictions used to design the nozzles and blades. Where the inlet nozzles are divergent, the steam admitted depends on throat area and initial pressure only. This is not true of convergent nozzles where, in addition, back pressure in the first-stage nozzles has an influence on steam passed. When the first stage is of the impulse type, the nozzle will practically always be divergent. The steam flow is then given by

$$w = \frac{50\,pA}{K} \text{ maximum lbs. steam passed per hr.}$$

p = Full throttle pressure, psi. abs.

A = Cross-sectional area of all first-stage nozzle throats, sq. in.

$K = \sqrt{\text{steam quality}}$,

or, if superheated,

$K = \sqrt{1 + .00065\Delta T}$,

in which ΔT = deg. of superheat.

The losses in a steam turbine, as is the case with the engine, are topped in magnitude by the heat in the exhaust. Unfortunately, this is difficult to reduce because of the troubles attending the use of steam of quality lower than about 85% in the turbine. Wet steam erodes turbine blades, due to the high velocity with which the particles of moisture strike them. Consequently, even if internal turbulence and friction could be avoided and ideal isentropic expansion be obtained, much latent heat of evaporation must be allowed to remain in the exhaust steam. The other losses which occur in steam turbines are:

1. Internal losses due to imperfect flow.
   a. Leakage. Past shaft gland packings, dummy piston, diaphragms and blade tips.

b. Blade, nozzle, and disk friction.
c. Throttling at the control valves.
d. Non-stream line flow at other than design conditions.

2. Mechanical losses.
a. Bearing and sealing gland friction.
b. Oil pump and governor power.

All of the internal losses of one stage are returned to the steam as it enters the next lower stage, so that one definite advantage of multi-staging is to

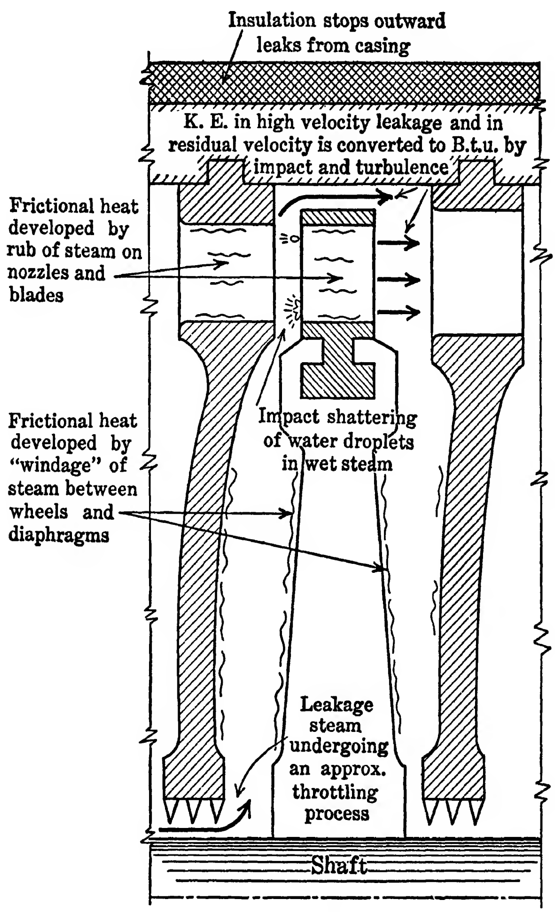


Fig. 14-33. Sources of inefficiency in a turbine stage. Though no heat is lost there, irreversible energy-degrading processes consume "availability."

make available to the next stage the thermal losses of the preceding one. This is one factor in the superior performance of multi-stage turbines. To illustrate the point, consider the case of friction on the nozzles. If the turbine casing is well insulated, the heat that is generated by friction of the steam against the nozzles simply elevates the nozzle temperature until it returns, by conduction

to the steam, as much heat as is generated by friction. Figure 14–33 shows how internal losses are reabsorbed by the steam as heat energy. The greater part of the stage loss is returned to the steam at the low pressure, and in turbine design it is considered that all of the reheat occurs at the low pressure of the stage. Figure 14–34 shows how an adiabatic heat drop, as in nozzles, followed by a reheat, in which the stage losses are added at the low pressure, results in a steadily increasing entropy in a multi-stage turbine. This is measured by *stage efficiency*, meaning the fraction of isentropic heat drop in a steam turbine stage that is transferred to the rotor as mechanical energy. The remainder goes to reheating the steam at the lower pressure. A well designed turbine may have a stage efficiency as high as 85%. This explains the typical expansion line of the steam turbine, which, as pressure is reduced, increases in entropy, whereas an ideal turbine would expand the steam at constant entropy. The increase of entropy corresponds to a decrease of energy availability, and that turbine which has the minimum entropy increase shows the best performance.

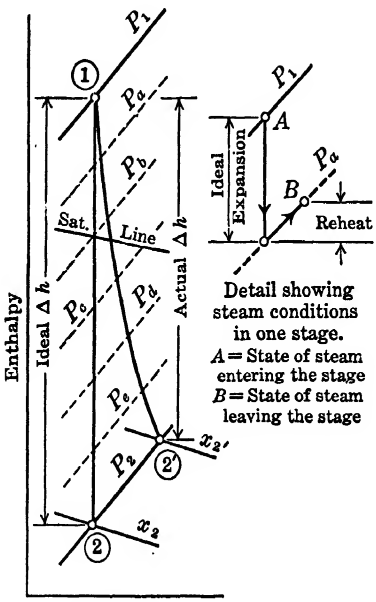


FIG. 14–34. Expansion of steam in a six-stage turbine traced on the h-s plane. 1–2 = Expansion without internal reheat. 1–2′ = Expansion with conditions depicted in Figure 14–33. P_a, P_b, etc. = Interstage pressures.

It is obvious from the foregoing that all of the heat taken out of the steam during its passage from the throttle to the exhaust must have been put onto the rotating shaft as mechanical energy. This is true because of the negligible radiation loss from the casing. Friction and power required to drive the governor and oil pump take their toll of this energy, but they rarely amount to as much as 5% of the decrease of enthalpy, practically all of which appears at the turbine shaft coupling as useful work.

Steam turbines are governor-regulated for constant speed in nearly all cases. The performance of a turbine is well shown if power output is used as the variable against which quantities such as thermal efficiency and steam rate are plotted. The steam rate curve data of Figure 14–35 would necessarily require a test to be performed to obtain it, but thermal efficiency is a derived quantity. Equations for the thermal efficiency, ideal cycle efficiency, and engine efficiency of a steam turbine are the same as those previously given for the steam engine. Turbine cycles always feature complete expansion. In ad-

dition to the thermal efficiency curve, the steam consumption (Willans line) curve can be derived from the steam rate. It has been found that this line is practically straight up to the operating point of maximum efficiency provided speed is constant.

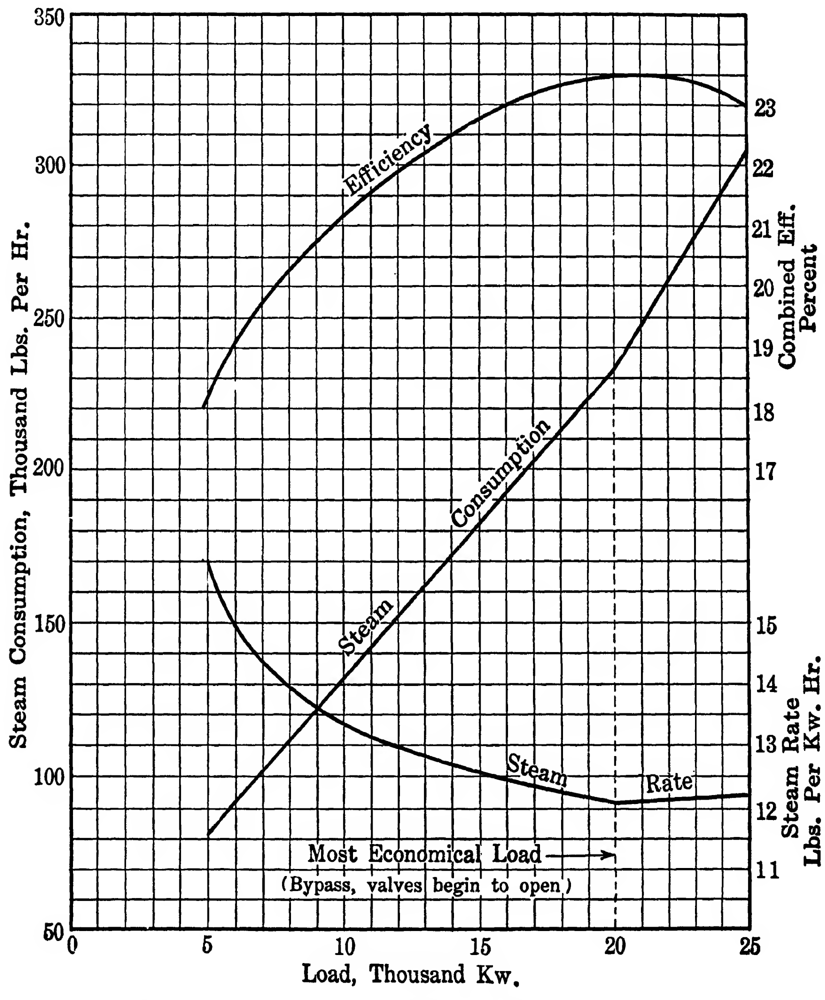


FIG. 14–35. Performance curves of a condensing steam turbo-generator unit. 190 lbs. per sq. in. gage 100° superheat; exhaust 1.5″ hg. abs.

Example 1: Illustrating the derivation of efficiency * and steam consumption lines from steam rate, take the steam rate at 8000 kw. load for Figure 14–35. For the steam condition stated, $h_1 = 1260$ B.t.u. $h_{f_2} = 60$ B.t.u.

* In the large turbo-generator field, engine efficiency is often given on an overall basis and includes the generator performance as well.

Then $\eta'_e = \dfrac{3412}{w'(h_1 - h_2)}$, with w' = lbs. steam per *kw. hr.*

$$\eta_t = \frac{3412}{14(1260 - 60)} = 20.3\%.$$

Steam consumption = 8000 × 14 = 112,000 lbs. per hr.

As a matter of fact, any two of these three performance curves are derivable from the other.

Example 2: Given initial pressure 200 psi., temperature 440°, exhaust 1 in. Hg abs. Engine efficiency 75%. Analyze the turbine performance to the extent permitted by these data.

By tracing an isentropic process on the Mollier Chart, beginning with 200 psi., ending at 1 in. Hg, the ideal heat release is determined to be 1236 − 853 = 383 B.t.u. Substituting in the equation for engine efficiency,

$$.75 = \frac{2545}{w \times 383}, \text{ whence } w = 8.85 \text{ lbs. per hp. hr.}$$

$$\text{Thermal efficiency, } \eta_t = \frac{2545}{8.85(1236 - 70)} = 24.7\%.$$

Heat remaining in exhaust steam = 1236 − .75 × 383 = 948 B.t.u. per lb.
Condition of exhaust for h = 948 B.t.u. and p = 1 in. Hg is 86% dry.

Example 3: The efficiency of a turbine stage is 78%. Steam enters the nozzles at 60 psi., 98% quality, leaves the stage at 35 psi. What is the gain in entropy caused by the losses?

The entropy of the entering steam is 1.619, its enthalpy 1158 B.t.u. Had the flow been ideal (streamlined and frictionless), the entropy leaving would still be 1.619, and the enthalpy 1118 B.t.u. The losses are 22% of the ideal heat release. .22(1158 − 1118) = 8.8 B.t.u. After absorbing this the steam is reheated to 1118 + 8.8 = 1126.8 B.t.u. This is the actual exit enthalpy. The chart shows that this enthalpy at 35 psi. pressure, requires the entropy to be 1.631. Gain of entropy = 1.631 − 1.619 = .012 units per lb.

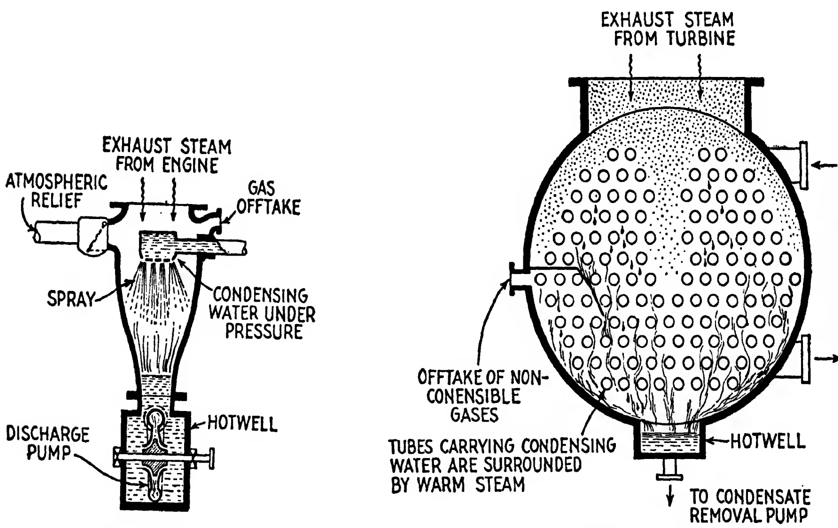
14–14. Extraction Turbine. A turbine is a steady flow machine. As long as it operates at one load condition the state of the steam passing any internal station in the casing remains nearly constant. So when steam of some particular pressure is wanted industrially or otherwise, and that state exists somewhere in the expanding steam of a turbine, a supply of it can be drawn out through the casing if an opening is provided at the proper point. This is called "extraction" or "bleeding." Regenerative cycle * power plants must use bleeder turbines. But *extraction* refers to any arrangement whereby steam is bled from a turbine at one or more pressures for any purpose whatsoever; i.e., feed-water heating, process steam, heating steam, etc. The terms "bled steam" and "extracted steam" may be used synonymously, as may also "bleeder point" and "extraction point."

There are two types of extraction, i.e., extraction at constant steam pressure, and extraction at whatever pressure exists in the turbine at the extrac-

* Described in Section 12–5.

tion point. Extraction at constant pressure requires that an extraction valve gear be provided to regulate the casing pressure at the extraction point. This is necessary because, not only would the extraction pressure vary with different amounts of extracted steam demanded, but varying loads on the turbine would cause the casing pressure at the extraction nozzle to vary. The extraction valve gear is often complicated by the use of a control or pilot valve to operate the main extraction valve. Turbines equipped with extraction valve gear are naturally more expensive than the simpler forms which have no pressure governing on the extraction lines. Industrial use of extracted steam often requires that the pressure of the bled steam be kept constant. Also, industrial use of the extraction turbine differs from central power station practice in that frequently a large portion of the total flow is extracted, whereas in the regenerative power plant only a small fraction of the total is used for feed-water heating. So central station turbines have simple extraction outlets if steam is bled for feed-water heating.

14–15. Condensers. A steam condenser is the device which accomplishes the act of reducing a vapor to a liquid through the extraction from it of the



(a) Contact type. (b) Surface type.

FIG. 14–36. Condenser action.

heat of evaporation it may contain. The condenser is an important part of the modern steam power plant, since large turbines are hardly ever non-condensing. It is placed close to the turbine and direct-connected to the exhaust opening. The condenser figuratively "swallows" the exhaust steam. The purpose of the steam condenser is to lower the back pressure on the prime mover, allowing the steam a larger pressure and temperature drop, thereby increasing both efficiency and capacity. A secondary purpose may be the col-

lection of condensate for boiler feed water. Condensers are applicable to both engines and turbines, although especially to the latter because their thermodynamic advantages occur chiefly in the low-pressure range. Engines may be operated condensing up to 26 in. Hg vacuum and turbines up to 29.5 in.

Condensers can be divided into *contact* and *surface* types. Steam and condensing water are intimately mixed in the contact type. Then the condensate and condensing water are withdrawn thoroughly mixed together. A dividing surface is interposed between steam and water in the surface condenser. Heat is transferred through this partition surface. The steam and condensing water never come into direct contact and are withdrawn from the condenser separately. The dividing surface is ordinarily a tube with condensing water circulating inside and steam outside. Elements of the jet condenser are: (1) nozzles or distributors for the condensing water, (2) steam inlet, (3) mixing chamber, (4) hotwell, (5) in some cases, a diffusing chamber or a tail pipe.

The elements of the surface condenser are:

1. Cooling surface for both air and steam, usually $\frac{3}{4}$-, $\frac{7}{8}$-, or 1-in. copper alloy tubes from 10 to 25 ft. long.
2. Tube sheets into which the tubes are expanded or packed.
3. Intermediate tube supports (sheets bored similar to the tube sheets but in which the tubes fit loosely).
4. Water boxes provided with circulating water connections, and enclosing the space furnished for water flow to and from the tubes.
5. Steam inlet, air and condensate outlet.
6. Hotwell in which the condensate collects.
7. Condenser shell enclosing and supporting the other elements.

Surface condensers are classified as horizontal or vertical by the position of their tubes. They are single-pass or double-pass according to whether the water passes the length of the condenser once or twice. They are also classified upon a basis of the shape of the shell, as, for instance, cylindrical, heart-shaped, U-shaped, oval.

Heat transfer in a surface condenser is a convection-conduction combination.

$$Q = UA\Delta T.$$

Q = Heat transferred per hour.

A = Heat-transfer surface.

ΔT = Mean temperature difference.

U = Overall heat-transfer coefficient.

Q is the heat necessary to secure the condensation of the prime mover exhaust steam. A is the (external) tubular surface, and ΔT should be the logarithmic value. Many different factors affect U in a complex manner so that it ranges from 500 to 700 B.t.u. per hr. per sq. ft. per deg. F.

Heat-transfer action in a surface condenser is hindered by the presence of non-condensable gases which mix with the film of condensate on the tube surface. The sources of air and other non-condensable gas leakage are

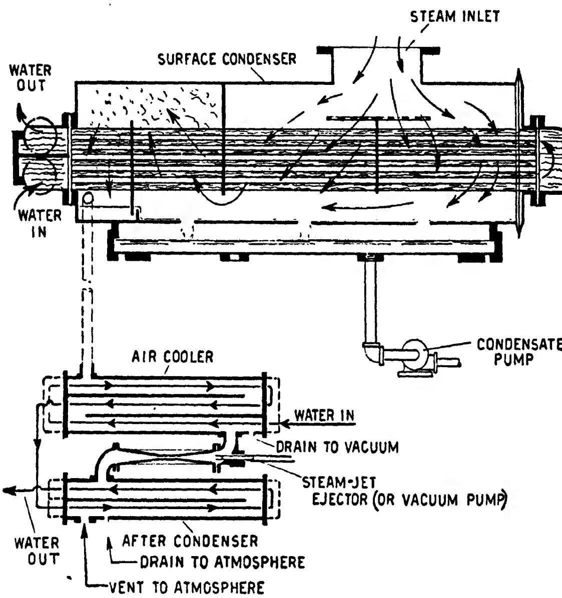


FIG. 14–37. Small surface condenser. Diagram showing tubes carrying condensing water, also means of withdrawing non-condensible gas. The air cooler reduces the volume required to be ejected against atmospheric pressure. (Courtesy Socony-Vacuum Oil Co.)

numerous. Some may come over with the boiler steam, or leak in through turbine packing gland or exhaust nozzle connection. Air leakage will seriously affect the heat transfer and every effort is made to minimize it.

The auxiliaries required for condensers can be arranged under two heads: first, those connected with the flow of water; second, those connected with the vacuum. The condensate pump serving a large condenser is usually of the centrifugal type but reciprocating pumps may be used for small condensers. The head on it is the vacuum plus friction of the piping to the storage tank plus the velocity head plus the difference in elevation between the discharge to the tank and the condenser hotwell.

A high-vacuum surface condenser requires around a hundred pounds of water per pound of steam condensed. Supply of circulating water is often a deciding factor in power plant location and a limiting factor in extension of existing plants. In case of limited supply of circulating water it may be

necessary to resort to cooling towers and ponds to cool the water for recirculation.

The auxiliary equipment having to do with vacuum includes the vacuum pump for removal of non-condensible gases, atmospheric relief, manometer,

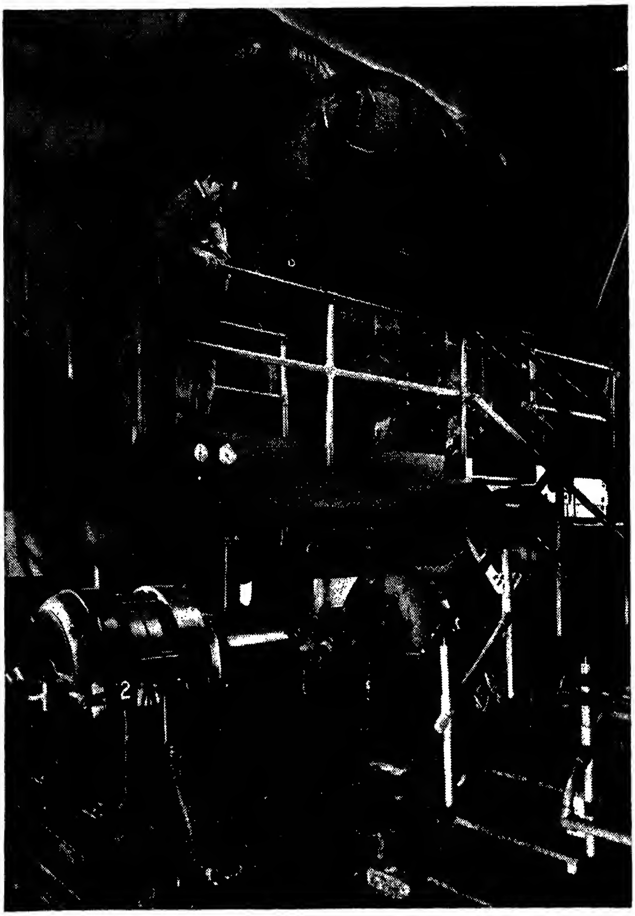


FIG. 14-38. Condenser and equipment. Showing one end of a large (about 20,000 kw. turbine) central-station type of steam condenser. End of a water-box shows here. One of the two circulating water pumps is seen. For position of a central station condenser relative to its turbine see Figure 14-16. (Courtesy Allis Chalmers Mfg. Co.)

and a device to break the vacuum if the condensate should abnormally accumulate in the shell.

PROBLEMS

1. A steam prime mover receives the steam at 150 psi. pressure, containing 2% moisture. What is the moisture content after an ideal working expansion to atmospheric pressure? How many foot-pounds of mechanical work are available per pound of steam? Solve by computation, using tables and check by use of Mollier Chart. Sketch a section of the chart (no scale), showing how graphical solution was made.

2. Repeat Problem 1 with changed steam conditions as follows: steam initially 250 psi., 500° F, exhaust pressure 1 in. Hg abs.

3. Steam at 200 psi., 400° F, if expanded isentropically to a low enough pressure, becomes wet. How low can the pressure be carried before the moisture exceeds 15%? What ideal quantity of work, in foot-pounds, is made available by this expansion? Solve by use of chart. Show, by sketch, how graphical solution was obtained.

4. Make a skeleton diagram of the engine shown in Figure 14–2, and label it to show the principal parts as listed on page 359.

5. Redraw Figure 14–5, making the length about 5 in. Then superimpose on it a crank-end diagram for an engine having events at same percentage of stroke as head end, and same clearance.

6. Construct sketches similar to Figure 14–4, but simplified, with valve and piston in position for (a) head-end compression, (b) crank-end cut-off.

7. Repeat Problem 6 for (a) head-end cut-off, (b) crank-end release.

8. Release is to occur at 90% of each piston stroke of a certain uniflow engine. The stroke is 18 in., clearance each end $\frac{3}{4}$ in. Find the required length of the piston and inside length of the cylinder.

9. Diagram an inside admission piston valve in position for head-end cut-off.

10. Repeat Problem 9 for head-end release.

11. Plot a Rankine engine cycle for the following data: Throttle pressure 100 psi., exhaust 14.7 psi. Per cent of stroke at events: cut-off 25%, release 95% of the forward stroke; compression 70%, admission 95% of the back stroke. Clearance 5%. Scales 1 in. = 20 psi., 1 in. = 20% of piston displacement.

12. Repeat Problem 11, but with an initial pressure of 125 psi., and cut-off 30%.

13. Draw timing diagram on a 4-in. diameter circle for head- and crank-end events as follows: cut-off 25%, release 90%, compression −30%, admission −5%, $R/C = 8$.

14. The angles, measured from HEDC, for events pictured in Figure 14–6, are:

CO_h	60°	CO_c	230°
C_c	110°	C_h	310°
R_h	150°	R_c	340°
A_c	170°	A_h	350°

The R/C ratio is 5. Make a graphical solution for and record the per cents of stroke of each event.

15. Draw the head-end Zeuner diagram (full scale) for engine data as follows: $\theta = 140°$, eccentric throw 3 in., valve outside admission, steam lap 2 in., exhaust lap $\frac{5}{8}$ in. Measure (from HEDC) and record the crank angles at the four head-end events—also the lead.

16. Draw the head-end Zeuner diagram (full scale) for the following outside admission engine data: Valve travel 5 in., steam lap 1.9 in., exhaust lap 0.45 in., lead 0.1 in. Record the angle of advance.

17. Given D valve with cut-off 35%, R/C of large magnitude, lead 0 in., valve travel 6 in., compression at 70% of the return stroke. Find the per cent stroke at release, and the laps. (Head end.) Full scale Zeuner diagram.

18. Assume that the answers to Problem 17 are $S = 2.4$ in. and $E = .7$ in. Construct to half scale a diagram similar to the central figure of 14–12. Ports are on 18-in. center lines.

19. Assume the following data for the engine of Figure 14–2: eccentricity at full load 2 in., exhaust lap .1 in., cut-off 30%, admission 2% BDC, R/C $4\frac{1}{2}$. Draw full scale head-end Zeuner diagram and find (1) per cent stroke at compression, (2) at release, and (3) θ.

20. After a steam engine (single cylinder) test the following data were available: bore × stroke × speed = 8 in. × 12 in. × 200 rpm. Piston rod 2 in. diameter, $p_1 = 150$ psi., $p_2 = 14.7$ psi., cut-off at 30% stroke, head-end indicator card 3.02 sq. in., crank-end 3.20 sq. in., scale of spring 80 lb., card length 3.15 in. What was the internal horsepower developed?

21. Data of Problem 20. What diagram factor is indicated by these data? For actual mean effective pressure take the average of head- and crank-end values.

22. A single-cylinder engine with 12 in. × 18 in. × 250 rpm. dimensions will be operated non-condensing, using steam at 75 psi. gage at the throttle. Cut-off 40% stroke. Using a diagram factor of 70%, and mechanical efficiency of 85%, estimate the brake horsepower available.

23. A non-condensing steam engine is tested at full load of 80 shaft horsepower and found to have a steam consumption of 2800 lbs. per hr. Throttle pressure 250 psi., quality 99%. Find (1) the actual thermal efficiency, (2) the "engine efficiency."

24. It is known that a certain type of engine will yield an "engine efficiency" of 70%. Estimate the steam rate of such an engine when working between terminal conditions of 150 psi., 100 superheat, and 26 in. Hg vacuum. Use Mollier Chart.

25. Re-solve Example 1, Section 14–9, for a blade circle diameter of 42 in.

26. Find the blade thrust, work per pound steam flow, and blade angle for an impulse turbine, $\alpha = 20°$, jet velocity 2500 ft. per sec. Tangential blade speed 850 ft. per sec. Solve for velocities graphically, using scale of 1 in. = 600 ft. per sec. Sketch the blade (2-in. width).

27. A single-stage impulse turbine supplied with steam at 125 psi. gage, dry and saturated, atmospheric exhaust, will drive an 1800-rpm. generator through a 4:1 reduction gear. Blade circle diameter 30 in., nozzle angle 18°. Draw the ideal vector diagrams, scale 1 in. = 600 ft. per sec. Find the efficiency of the blading and the work done per lb. steam flow. Neglecting all friction and electrical losses, what steam flow is required per kilowatt output?

28. Steam is to be used most efficiently (i.e., residual velocity wholly axial in direction) in the following classes of ideal turbine staging. If $u = Kv_2$ for a plain single-stage impulse turbine meeting the above specification, what is the corresponding multiplier for:

a. Four-stage Rateau staging.
b. Ten-stage Rateau staging.
c. Single-stage two-velocity Curtis staging.
d. Four-stage each being two-velocity Curtis staging.

29. Using the instructions of Problem 28, find the multiplication for:

a. Six-stage Rateau staging.
b. Six stages, each being two-velocity Curtis staging.
c. Three stages, each being three-velocity Curtis staging.

30. How many stages should an ideal Rateau turbine have in order to use steam most efficiently between 150 psi., 100° superheat, and 26-in. vacuum? Blade speed 600 ft. per sec. 20° nozzles.

31. Trace the expansion of steam through an ideal three-stage (Rateau) turbine by use of the Mollier Chart. Initial state 115 psi., dry and saturated, exhaust 15 psi. Equal heat drops per stage. Record on a no-scale sketch of the chart the intermediate pressures, the stage heat release, the initial state, and the final state.

32. An impulse turbine blade moving at 500 ft. per sec. is symmetrical having lip angle of 40°. It receives steam smoothly at a relative velocity of 1750 ft. per sec. What thrust is exerted on the blade per pound per second steam flow over it?

33. Solve Problem 32 had the exit angle been 50° instead of 40°.

34. A reaction moving blade row has β_1 70°, β_2 20°, and 5 B.t.u. heat drop in the steam passing it. $u = 450$ ft. per sec. $v_{r_1} = 400$ ft. per sec. Using scale of 1 in. = 100 ft. per sec., find thrust per pound per second.

35. Same data as Problem 34, except $u = 500$ ft. per sec., and $\Delta h = 6$ B.t.u.

36. An impulse turbine has four divergent nozzles in the first stage, these having $\frac{3}{16}$-in. diameter throats. Throttle steam conditions 400 psi., 500° F. How much steam will this turbine use per hour when all four nozzles are in operation?

37. A 20,000-kw. steam turbine might be expected to have a steam rate of 12 lbs. per kw. hr. What throat area needs to be incorporated into the divergent first stage nozzles? Steam pressure 600 psi., temperature 750° F.

38. A steam turbine has a "Willans" line described by the equation $w = 5500 + 7.5P$, w being hourly steam consumption, P being horsepower output. The most economical load is 10,000 hp., after which the 7.5 is increased 10% per 1000 hp. Draw the steam consumption graph between no load and 12,500 hp. Scales 1 in. = 40,000 lbs. per hr., 1 in. = 2000 hp. abscissa. Derive, and plot on the same abscissa, a steam rate curve. Ordinate scale 1 in. = 4 lbs. per hp. hr.

39. The hourly steam consumption of a certain turbo-alternator is given by the equation $w = 20{,}000 + 8.5\,P$, where w is pounds, and P is kilowatts output. What is the overall efficiency at (a) 10,000, (b) 25,000 kw. load? Steam conditions: initial, 500 psi., 600° F, final 1 in. Hg.

40. A turbine has steam rate of 10.5 lbs. per hp. per hr. at a certain load. Steam initially at 250 psi., 100° superheat, exhaust condition 2 in. Hg, 86% quality. Find η_e and η_t for this load.

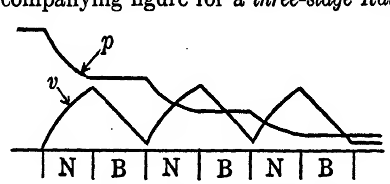
41. The steam generator of a turbine plant produces 350 psi. steam with 250° superheat. Condenser vacuum 27 in. Engine efficiency 75%. Estimate the thermal efficiency, and the enthalpy of the exhaust steam.

42. Steam enters a certain turbine stage at 40 psi., 92% quality, leaves at 20 psi., 90% quality. Plot the condition of one pound of the steam in this stage on h–s axes. 1 in. = 20 B.t.u., 1 in. = .05 B.t.u. per deg. Calculate the stage efficiency represented by these data.

43. At the entrance to a turbine stage steam has a pressure of 10 psi., quality 90%. Leaving the stage the entropy is 1.68, and the pressure 3 psi. Find (1) work produced by this stage, (2) stage efficiency.

44. A steam turbine is thought to have an engine efficiency of 70%. It receives steam at 350 psi., 500° F. What maximum condenser vacuum can be carried without exhaust steam quality becoming less than 85%?

45. Typical variation of steam pressure and velocity axially in a multi-stage turbine is shown in the accompanying figure for a *three-stage Rateau turbine.* N = end-



wise space occupied by nozzles, B by blades. Draw a similar graph, no scale, for a three-stage Curtis type turbine where each stage is of the two-velocity type.

46. Same ideas as Problem 45, except turbine type is to be ten-stage Parsons.

47. Same ideas as Problem 45, except turbine type is to be one two-velocity Curtis stage followed by four Rateau stages.

48. 45,000 lbs. of steam per hr. at 2 in. Hg and 88% quality leave a turbine at full load. How many gallons of condensing water must be pumped through the condenser per minute in order to maintain that pressure? Water temperature in 60° F, out 90° F. Assume condensate at saturation temperature.

49. 10 lbs. of steam at 1 in. Hg, 86% dry, are condensed per hr. by each square foot of condenser surface. Water temperature in 65° F, out 75° F. Calculate the coefficient of heat transfer U.

50. A steam condenser which is to condense 25,000 lbs. of steam per hr. at 1 in. Hg, having 995 B.t.u. per lb., will be equipped with 1-in. diameter tubes. Condensing water will enter at 60° F, leave 5° less than steam temperature. Tubes are 18 ft. long. Given $U = 650$ B.t.u., find (a) number of tubes required, (b) gallons per minute, condensing water.

CHAPTER 15

Conclusion

15-1. Auxiliary Equipment. In these pages we have attempted to show how energy can be transferred and transformed. Primarily, attention has been given the problem of converting the potential heat energy of a fuel into mechanical work. None of the prime movers considered can function without the services of some auxiliary equipment. The automatic types of I.C. engines are the most nearly self-sufficient, but even they will be found with exhaust mufflers, fuel pumps, starting systems, air filters, and other extraneous equipment. The heart of the steam power plant is the steam generator-prime mover group, but a wealth of other equipment is used either because it furnishes a vital service to the main unit, or because its presence results in better performance than could be had otherwise. A host of auxiliary and accessory equipment has been developed, much of which is of great ingenuity and technical interest. However, we would unduly extend this survey of applied energy if more than a cursory classification and identification of such were attempted. We venture to make this distinction between auxiliary and accessory equipment. An auxiliary piece of equipment is one without which the main unit could not function normally—or possibly at all. An accessory is something which, although usually very beneficial and desirable, is not absolutely vital to the functioning of the main unit. So, an exhaust muffler is an accessory; a carburetor an auxiliary. A boiler feed pump is a steam plant auxiliary; an automatic feed-water regulator an accessory. This distinction probably will not enjoy general acceptance, but these remarks will have accomplished their purpose if they result in less indiscriminate usage of *accessory* and *auxiliary*. Mostly, auxiliaries were considered along with the prime mover itself—except for the steam generator, whose service equipment is so extensive that much of it has been relegated to this spot. A reasonably complete list of steam plant equipment would include:

- Secondary heating surfaces.
  - Air preheaters.
  - Economizers.
- Feed-water conditioning equipment.
  - Pressurizing.
  - Purification.
  - Heating.
  - Flow control.
- Draft systems.
  - Fans.
  - Dampers.
  - Controls.
  - Chimneys.
  - Ducts.

Fuel systems.
- Conveyors.
- Storage.
- Crushers, screens, etc.
- Combustion control.

Piping for flow of fluids.
- Pipes and fittings.
- Valves.
- Drainage and support.
- Expansion joints and bends.
- Insulation.

Condensing water.
- Pumps.
- Conduits.
- Screens.

Soot and ash disposal.
- Conveyors.
- Storage.
- Quenching.
- Soot blowers.

Electrical installations.*
- Generators.
- Voltage transformers.
- Circuits.
- Protection.
- Motors.

Instruments.

15–2. Secondary Heating Surfaces. Absorption of the heat of combustion takes place mainly in the boiler. Sometimes combustion air heaters and feed-water heaters are included in the path of the products of combustion. These surfaces are cheaper to build than boiler surfaces. Their presence permits high boiler outlet temperatures to be carried without incurring excessive heat losses, so increasing the rate of heat transfer through boiler heating surface.

The air preheater is installed between the boiler flue gas outlet and the chimney. The heating surface is composed either of tubes with flue gas inside and the air to be heated outside, or of rectangular plates spaced about one inch apart, leaving alternate gas and air passages. Its use is chiefly justified on economic grounds.

Two principles are employed for heat transfer in air preheaters. The recuperative principle implies transfer of heat through a separating partition, such as the walls of a tube, by continuously recuperating the cool side with conduction of heat from the hot side. Regenerative heaters are those which alternately heat and cool the same mass, regenerating it thermally by passing hot spent gas over its surface. As the process is one of exchange of heat with little or none being lost, the temperature changes of the flue gas and air, their specific heats, and flows are related in this way,

$$w_a c_a \Delta T_a = w_g c_g \Delta T_g.$$

If all the air used for combustion is preheated, then $w_g = w_a +$ weight of fuel burned with w_a.

The economizer of a steam power plant is a heat-exchange surface the purpose of which is to recover waste heat in the flue gas by absorbing it in the boiler feed water. Such economizers often form an integral part of the

* The author supposes that including the highly important electrical end of the steam power plant in a list of auxiliary and accessory equipment will get a poor reception from the electrical fraternity. But indeed such is extraneous to the prime mover. Also, steam plants will be found having no electrical installations, i.e., the locomotive.

boiler surface, with heating surface in the form of tubes. Where a separate unit, it is located between boiler outlet and preheater. Heat is recovered from flue gas by passing it through the tube surface into the feed-water stream, which circulates inside the tubes. Ordinarily, no steam is produced

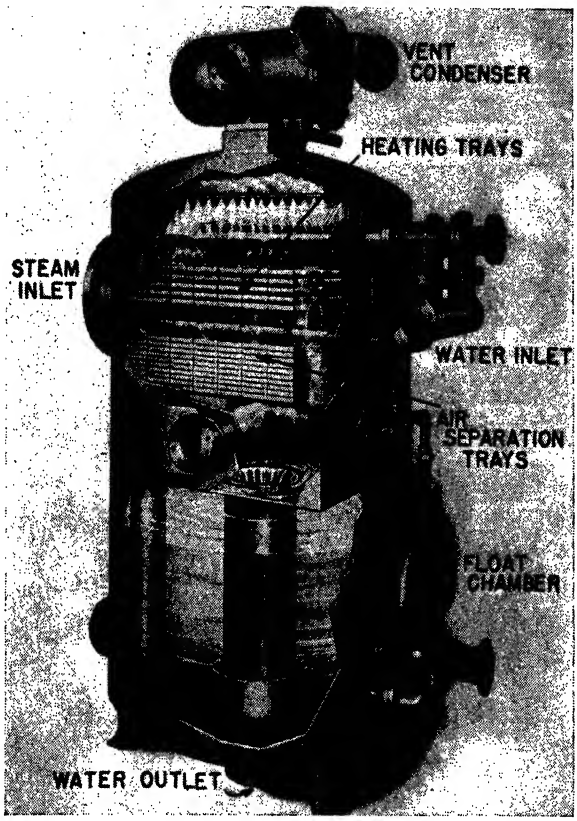


FIG. 15-1. Deaerating type contact feedwater heater. (Courtesy Cochrane Corp.)

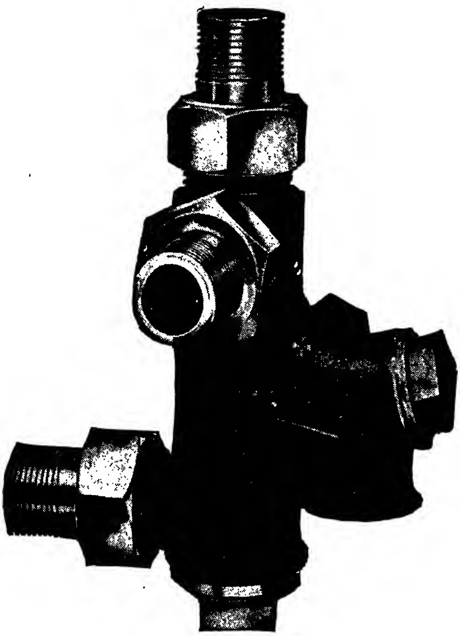
in the economizer. The heat transfer is represented in a manner similar to the air preheater.

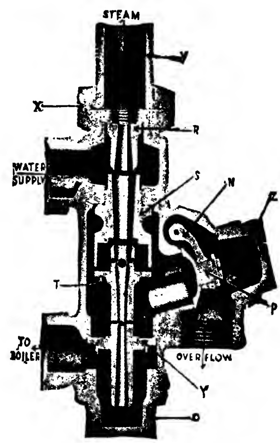
$$w_w c_w \Delta T_w = w_g c_g \Delta T_g.$$

If the evaporation per pound of fuel is w'_w, and the air-fuel ratio w'_a, then w_g/w_w for the above equation becomes $(w'_a + 1)/w'_w$.

15-3. Feed-Water Conditioning. Water, to be suitable for boiler feed, should have the requisite degree of purity, be under sufficient pressure to flow in against steam pressure, and be regulated in flow so as to compensate for the water removed from the boiler by evaporation. In addition, the economic advantages of some preheating are so considerable that feed-water heating is common practice. The equipment in this category includes feed-water pumps, feed-water heaters, feed-water treatment, and feed-water regulation.

Almost every kind of water pump at some time or other, has been employed for boiler feeding. Section 15–7 is devoted to water pumps. Here we mention two types whose application is almost entirely for boiler feeding, viz., (1) the injector, (2) the direct-acting steam pump. The *injector* is a jet-type feed pump, limited to regular service on small boilers and stand-by





SECTION

P—Overflow valve
R—Steam jet
S—Combining tube
T—Ring valve
Y—Delivery tube

Fig. 15–2. Automatic feed-water injector. When steam at boiler pressure is admitted, a high velocity jet rushes through the combining tube and out the overflow via the ring valve and the spill holes in the delivery tube. This creates a suction capable of drawing water through the supply pipe so that it enters and mixes with the steam in the combining tube. The condensation creates a partial vacuum, closing the ring valve. Although water still escapes through the spill holes, the overflow decreases as steam and water reach the right proportions for full pressure rise in the diffusing delivery tube. The delivery soon builds up a pressure exceeding boiler pressure and water flows to the boiler past a check valve (not shown) in the feed line. Flow in the overflow now tends to reverse but only results in tight closure of the hinged overflow valve. If the flow of water should be suddenly broken from any cause, the automatic feature will cause immediate restarting without manual attention. (Courtesy Penberthy Injector Co.)

service on small- and medium-sized boilers. Like centrifugal apparatus, the injector operates on the principle of a velocity-pressure conversion but differs from the centrifugal in the manner of creating the velocity. The water acquires its velocity by impact with high-velocity steam leaving a nozzle, after which pressure is built up in a diffuser. The injector is simple, compact, inexpensive, and with no moving parts to wear or require adjustment. As a combined feed-water heater and pump its efficiency is high; but, as a pump alone, very low (less than 5%). Its characteristics recommend it to locomotive

service but not to regular service in an efficient stationary plant where feed water is otherwise heated.

The direct-acting pump is a steam-driven pump of the piston and cylinder type, not having crankshafts, flywheels, or similar rotative apparatus. The kind most often found has two steam cylinders arranged side by side. The water cylinders are also two in number, arranged side by side. The *duplex*

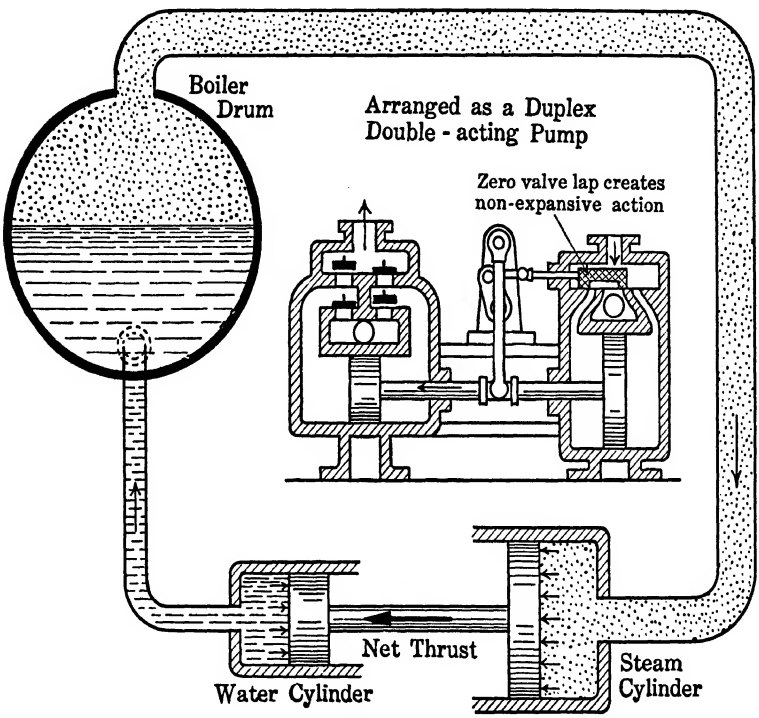


FIG. 15-3. Principle of the direct-acting, reciprocating steam pump. Although the unit pressures of the fluids in the two cylinders are substantially equal, a decisive net thrust is developed on account of the differential piston diameters.

pump is in reality a twin-cylinder pump. The special advantage of the twin arrangement comes from the convenience of valve operation in a twin-cylindered pump. The steam valve of one of the cylinders is caused to reciprocate properly in its valve chest with a motion derived from the travel of the piston rod of the other.

In order to insure ample operating pressure, the ratio of steam to water cylinder diameters is made large enough to include considerable margin of reserve, being about 1.6 for ordinary boiler feed service. Control of capacity is exercised through speed variation by throttling the steam line.

The direct-acting steam pump consumes from 100 to 300 lbs. of steam per hp. hr. Thermal efficiency is so low as to have no comparative meaning and

in its place is substituted *pump duty*; that is, the foot-pounds work done in the pump cylinders per million B.t.u. chargeable to the steam end. The high steam consumption is caused by non-expansive use of the steam. Were the steam expanded the pump would stall before reaching the end of its stroke.

Feed-water heaters are divided into two classes—the contact and the surface heaters. Economizer surface and a portion of the boiler surface are actual water heating surface; however, it is customary to refer only to equipment obtaining heat from steam as feed-water heaters.

The common contact-type heater is called an "open" heater because the interior is open to the atmosphere through the vent pipe. The *open heater* is ordinarily built up in rectangular form, but heaters for other than atmospheric pressure are constructed in cylindrical form of cast iron or steel plate. The open heater is provided with tiers of trays, properly perforated and inclined to break up the flow of water, delivered by gravity from a distributing trough, into a multitude of small cascading streams which present a large surface to the steam. It is possible to heat water to the temperature of saturated steam entering the heater if there are no non-condensable vapors. Heating is by direct conduction from steam to water.

The surface heaters are divided into steam tube and water tube types. Most heaters are of the water tube type. These heaters can also be divided into straight tube and bent tube (U tubes and steam coils), and into single- or multi-pass. The surface heater is used when water is to be heated under pressure without direct contact with the steam. A typical surface heater would be similar to Figure 4–3 except for the absence of internal baffles. Steam flows in only, the outflow being condensate drained from the bottom of the heater. Figures 12–5 and 12–7 show how heaters may be used in a feed-water system.

Example 1: Steam engine exhaust is mixed with feed water in an open heater, thus heating the mixture to 210° F. Incoming feed water 65° F, incoming steam 14.7 psi., 89% dry. The boiler generates steam to keep the engine supplied. We will estimate what part of the engine exhaust is saved by the heater.

Basing calculations on one pound of feed water equalling one pound of steam to the engine, action at the heater may be described as follows:

Let x = fraction of a pound of exhaust steam mixed with and condensed by the 60° water; h its enthalpy. Then $1 - x$ will be wasted to the atmosphere, and $1 - x$ lbs. of make-up water at 60° will replace it.

In the heater:

$$(1 - x)(h_{f_{60}}) + xh = (1 - x + x)h_{f_{210}}.$$

$$(1 - x)28 + x(180 + .89 \times 970.3) = 178.$$

$$x = 0.15 \text{ lb}.$$

This computation shows that a comparatively small part of the heat in the exhaust is saved by the heater when the steam is generated exclusively for

the use of the engine. However, if the engine were to be using only 15% of the boiler output (remainder to heating, industrial processes, condensing turbine, etc.) then all the exhaust heat would be salvageable by the heater.

Example 2: The flow diagram of Figure 12–7 includes a surface heater. Assume that the flow of feed water through this heater is 32,000 lbs. per hr., and that enough steam is to be extracted to raise the feed-water temperature from 92° to 250° in this heater.

This example shows how the quantity of steam required is computed, knowing the state of the steam extracted from the turbine.

To heat the water to 250°, the steam would have to be somewhat hotter, say 260°. This fixes the minimum pressure at which the steam could be extracted. The saturation pressure at 260° is 35.4 psi. Assume the steam to be 95% dry at that pressure, and that the condensate drained from the heater is at 255°.* Then $h = 228.7 + .95 \times 938.6 = 1120.3$ B.t.u., h_f at 255° $= 223.6$ B.t.u.

Let x = weight of steam used for heating, lbs. per hr. The following computation equates the heat energy released by the steam in condensing to the heat energy absorbed by the feed water.

$$(1120.3 - 223.6)x = (218.5 - 60.0)32{,}000.$$

$$x = 5750 \text{ lbs. per hr.}$$

The purpose of *water treatment* is to prevent the formation of scale on the water side of the boiler, to reduce corrosion of boiler metal to a minimum, and to insure the absence of foaming and priming under all conditions of operation. The higher the rate of heat transfer per square foot of surface, the more important it becomes to keep that surface scale-free, because the scale can both reduce the steaming capacity and cause overheating of the tubes.

Natural waters usually contain dissolved salts and gases, also some organic and inorganic material in suspension. They rarely are neutral in reaction. The troubles caused by the feeding of water of undesirable quality are scaling, corrosion, foaming and priming, and embrittlement. The solubility of salts in water varies with the temperature of the water solution. Some salts are much more soluble in hot water than in cold water, whereas the solubility of other salts decreases with increasing temperature. The latter salts form scale. These salts may be relatively soluble in the cooler parts of the boiler, but in the boiler tubes, because of the higher temperature, the saturation point may be reached although the actual concentration is quite low. The scale is deposited directly on the metal since the hottest water is that which is in contact with the metal surface. The hazard of scale lies in tube failures. Furnace temperatures are far higher than can be withstood by uncooled tubes. A layer of scale on the tube surface has the effect of preventing normal heat

* Surface heater condensate temperatures are uncertain because of the random fashion by which it reaches the hotwell. It can approach the steam temperature, 260° F, or might be as low as the outgoing water temperature, 250° F.

transfer through the tube wall to the boiler water and thereby increasing the temperature of the tube metal, causing wasting away on the fireside by oxidation. A very slight layer of scale is sufficient to raise the tube-wall temperature enough to cause failure.

Corrosion of the water side of boilers may occur in three ways: by acid attack, by electrolytic dissolution, or by oxidation. Combinations of these three processes also are possible. Direct oxidation of boiler metal results from the reaction of either dissolved oxygen or oxygen from water molecules with the iron of the boiler tubes and shells. Dissolved oxygen which enters with the boiler feed water is a major cause of boiler corrosion.

Natural water may receive thermal or chemical treatment in order to rid it of harmful impurities. Thermal treatment could consist of a distillation of the make-up water, the distillate being, of course, pure neutral H_2O. Chemical treatment is classed as external or internal, depending on whether the reactions are completed before the water enters the boiler or in the boiler. Internal treatment, if scientifically designed and controlled, is an effective method.

External *water softeners* are of two types—precipitation and base exchange. A precipitation softener embodies the principle of using calculated quantities of soluble reagents to react with the hardness in the raw water. Two treating tanks are used in the intermittent system, one supplying treated water to feed service, while the other is receiving its charge of chemicals and water or maintaining a quiescent condition so that the precipitate may settle out. Water flows continuously from inlet to outlet in the continuous type of softener. Reagents are added at the inlet and what precipitate does not settle out in the reaction tank is removed by filtration. A base exchange softener removes the hardness by a simple filtration of the water through a bed of active material which exchanges its sodium base for the scale-forming magnesium and calcium in the water. Natural and artificial zeolites are used as the active material.

Practically all chemical treatment tends to increase the concentration of soluble salts in the boiler water. These may be harmless until they completely saturate the water, then scale can be formed. Judicious removal and wastage of some quantity of well-saturated (chemically) boiler water will hold the concentrates under control. This is *blow down*, usually accomplished by opening the drain, or blow-off, valve a minute or two at suitable intervals, say daily. If steadily and slowly carried out, the action is known as *deconcentration*.

Evaporators are used to produce pure water from sea water or other impure source of supply, using live steam to produce the vapor. Evaporators are classed as film-, flash-, or submerged-tube types. The first and last are steam-tube types; in the former the raw water trickles over the hot tubes, in the latter the tubes are entirely surrounded by the water being evaporated. The flash type produces steam by dropping the pressure on water at the saturation temperature.

An evaporator system may be single effect, in which the vapor is produced from one evaporator, or multiple effect, in which the vapor is produced from several evaporators in series. In a multiple effect system the vapor from one evaporator becomes the heating steam in the succeeding. Unusual conditions met in industrial or steam heating plants may require so large a fraction of make-up as to warrant double, triple, or quadruple effect evaporators.

Deaeration is practiced when the gas that water contains would have undesirable effects. Often dissolved oxygen is objectionable because of its corrosive action. This is true in the case of the high-pressure steam boiler, where a small amount of oxygen dissolved in the feed water may become quite active in attacking the boiler metal under the high pressure and temperature conditions there experienced. Steam boiler operators often treat their boiler feed water in *deaerators* to remove this oxygen. Deaerating action in a thermal type deaerator is obtained by first reducing the solubility of the gas through heating the water (under pressure); second, reducing the pressure and producing explosive boiling; and third, controlling the agitation of the water subsequent to the second action in a partially evacuated region.

Disastrous consequences can follow extreme low water in a boiler. Also damage can be done to prime movers if sufficient water is inadvertently pumped into the boiler to fill it to overflowing. The ideal water level is about midway of the steam drum of a water-tube boiler. High and low water alarms are usually included in the water column. Furthermore, fusible plugs are screwed through the shell or drum at the minimum safe low water line. When these are no longer covered by water their composition core melts out and steam blows violently into the furnace. This serves as a warning, a furnace cooler, and a pressure relief.

A *feed-water regulator* is the "governor" of the feed-water system. Under modern conditions it is necessary for feed water to flow into the boiler almost as rapidly as the steam flows out—and since boilers are approaching the flash type, it is plain that the feed-water regulation should be automatic, purely a machine function. It cannot be done very successfully by hand. All regulators operate sensitive to water level. They may be classified as: (1) thermostatic (*a*. mechanical, *b*. vapor pressure), and (2) float-operated. Control is exercised on feed-water inflow by one of the following methods:

1. The regulator action varies the steam to a steam-driven feed-water pump.
2. The regulator action controls a valve between the boiler and the discharge of a constant speed feed-water pump, motor- or turbine-driven. Alternately the control valve is located in a by-pass line. When it is opened flow to the boiler decreases.
3. The regulator operates an electrical switch to start or stop the feed-water pump motor.

Of the above, (1) and (2) cause variable but continuous flow, while (3) gives intermittent flow. Figure 15–4 shows a thermostatic tube regulator. The element is a long inclined tube, one end of which is below the normal water level, the other above. The tube is connected so that water stands in it at the same level as in the boiler. If the boiler water level changes, so does the length of tube in contact with the relatively cool water. Thereupon the tube changes in length by thermal action. This expansion is mechanically multiplied and used to adjust the control valve.

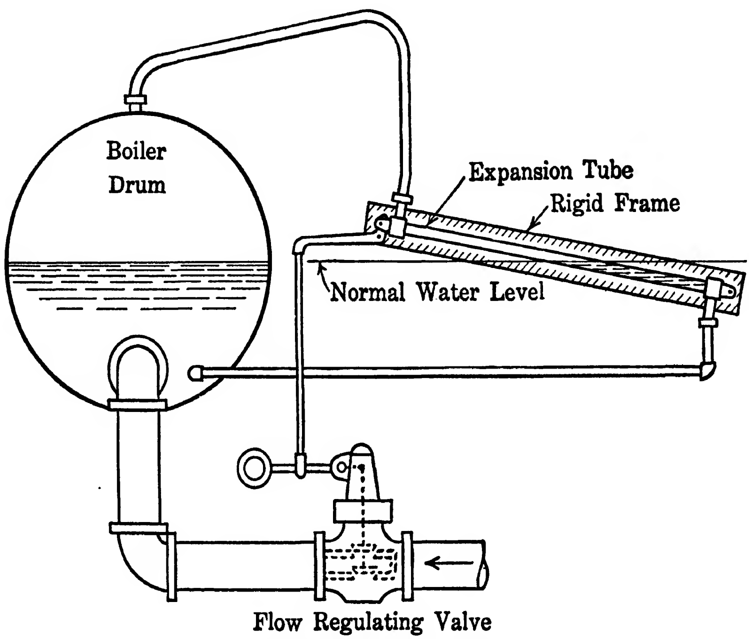


FIG. 15–4. Feed-water regulation by thermal expansion of a mechanical element.

15–4. Combustion Control. Good combustion of fuel in a furnace, efficiently, smokelessly, and at the desired rate, is difficult to achieve by manual adjustments if the steaming rate varies. Systems of automatic combustion control have been developed which, when installed, relieve the furnace operator of the need of making some or all of those adjustments, depending on the degree of automatic control supplied.

The steam boiler furnace furnishes an example of as complex a combustion control requirement as any to be found. The control consists of simultaneous and proportionate variation of fuel and air to meet the load and is exercised through such actions as changes of fan speed, damper positions, and fuel feed. Steam pressure is used to energize automatic control systems. If combustion is excessive, more steam will be generated than is used. The surplus, accumulating in the boiler, tends to raise the pressure. Then the master element of

the regulator senses the change and translates it into reductions of fuel and air to the firing equipment. Combustion control is of special benefit in plants where the highly variable demand makes it next to impossible to maintain a

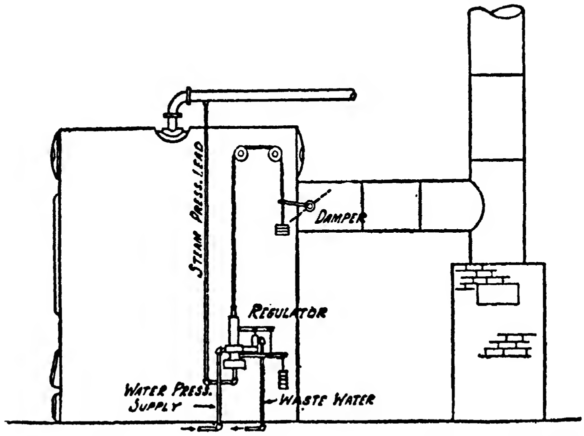


FIG. 15–5. Elementary combustion control by damper regulator.

uniform steam pressure manually. Furthermore the excess air can be more readily maintained at the best value. The ideal combustion control equipment should accomplish the following objectives:

1. Regulate rate of firing in accordance with load.
2. Maintain steam pressure within predetermined limits.
3. Maintain proper proportioning of fuel to air.
4. Maintain a slight furnace vacuum.

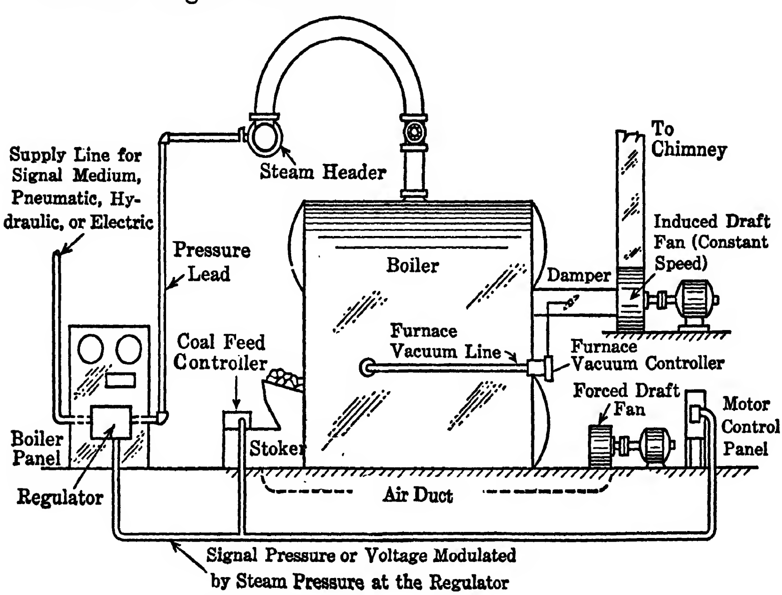


FIG. 15–6. Combustion control system.

A simple form of control wherein steam pressure automatically adjusts the draft is applicable to the smallest plants. The damper regulator substitutes mechanical control for the frequent opening and closing of dampers by firemen. The rate of fuel feed remains under manual control.

A complete combustion control system is also illustrated. The steam pressure is transmitted to a master regulator which modifies the signal supply medium to a variable signal which may be pressure or voltage. The individual controllers upon receiving this signal act to change motor speed, damper opening, coal feed, etc. During the process of initial adjustment of the system the controllers have been corrected so that the signal transmitted by the master regulator results in just the change in motor speed, damper position, etc., needed by the boiler unit at that particular rate of combustion. The signalling medium may be air under either plenum or vacuum, a liquid, or electricity. The furnace pressure controller is independent of steam pressure, for a predetermined vacuum should be maintained in the furnace at all ratings.

15-5. Soot and Ash. Soot and ash may be considered by-products of combustion. Ash appears in conjunction with coal fires, but soot can deposit with any fuel. Soot tends to collect on the cooler boiler tubes, the economizer, and the air preheater surfaces. It seriously retards heat transfer and should not be left to accumulate. Removal of soot is accomplished by steam jets produced either from hand-operated lances poked through cleaning openings in the setting, or by a built-in *soot blower*. Figure 15-7 shows a built-in type cleaning a section of finned economizer tubes. The steam is turned into the

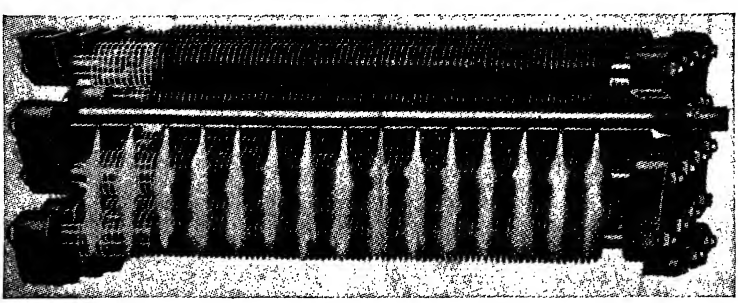


FIG. 15-7. Steam jet soot blowers in action, cleaning outside of a finned-tube economizer.

blower tube, which is then rotated, causing its multiple nozzles to blast the soot from all tubes located near it. Several such elements may have to be used in a large steam generator.

Since all coal has ash, and ash is incombustible, the disposal of ashes is a necessary service. Combustion of coal in a furnace results in the accumulation in an ash hopper, or ash pit, of the refuse, which, with proper handling of combustion, will be principally ash, but may in some cases contain up to 20%,

by weight, of carbon in the unburned state. All the ash should reach the ash pit or hopper, but as a matter of fact, from 5% to 40% may leave the furnace in the flue gases—carried in suspension. This "fly ash" content can reach nuisance levels where the coal is fired by sprinkler stokers or in pulverized form. The large amount of coal consumed by public utility plants is potentially capable of discharging from the stack so much ash to the surrounding territory that an endeavor is made to reduce the percentage of ash leaving with the gas, and to capture the residual ash from the gas. Water sprays, density separation, and electrostatic precipitation are possible methods.

Removal of ashes, and their disposal, is no simple problem because, first, the ash is *dusty*, hence irritating and annoying to handle; second, it may contain *clinkers* which must be broken; and third, it is *abrasive* and will wear all conveyor parts in contact with it if there is any relative motion. Ash disposal systems consist of some means of removing ash from the furnace, loading it on a conveyor system, unloading from the conveyor to storage, and a means of disposing of stored ash. Ashes can be raked from ash pits to boiler room floors, and then shoveled into wheelbarrows or cars, or raked to gratings where they will fall into a conveying system. The large furnace will often be designed so that the ash may be handled by gravity directly from the hoppers to cars or conveying system. Present-day conveying systems variously employ bucket conveyors, scrapers, pneumatic conveyors, steam jets, and flushing water jets.

15–6. Draft Systems. Draft is a pressure-differential that operates to move gases. Combustion requires air. To move this air through the fuel bed and to produce a flow of the gaseous products of combustion out of the furnace, then through the boiler, economizer, etc., requires a difference of pressure equal to that necessary to accelerate the gases to their final velocity, plus friction losses. This difference of pressure is called draft whether measured above or below atmospheric pressure. The pressures are so small that they must be measured by manometers reading in inches of water.

Draft can be obtained by use of chimneys, fans, steam or air jets, or combinations of these. The chimney is probably the most common.

A *chimney* is a vertical tubular structure of masonry, steel, or reinforced concrete, built for the purpose of enclosing a column of hot gas, to produce thereby a draft. In addition to the useful draft it produces, a chimney must also overcome the friction loss in the chimney itself. Ordinarily, the economic chimney gas velocities range between 20 and 40 ft. per sec. The flue gas flow equals cross-sectional area of the chimney times the gas velocity.

A chimney produces a draft by a simple principle of thermodynamics. When gas is heated it expands in volume and decreases in density, in which condition it may be displaced by a more dense gas. The light, hot flue gas is confined by the chimney. The tendency of hot gas to move up the chimney

is proportional to the height of the stack, since the difference of weight of equivalent columns of air and flue gas is greater the higher the columns.

The draft of a stack is, in an elementary way, expressed by:

$$P = H(d_a - d_g).$$

P = Draft, lbs. per sq. ft.

H = Chimney height, ft.

d_a, d_g = Atmospheric and chimney gas densities, lbs. per cu. ft.

Many empirical formulas are advanced for computing the height of a chimney required for a given boiler. A rational scientific approach would necessarily be based on the above equation, as it truly represents the physical action actually creating the draft.

Mechanical draft may be classified as *forced* or *induced*, the former having the combustion air placed under a plenum, the latter referring to gas movement into a region of partial vacuum. With forced draft alone, furnace gases seep outward through cracks in walls, and blow through opened doors and ports. Induced draft alone allows considerable undesirable dilution of the products of combustion unless furnace, casings, ducts, etc., are maintained air-tight. A logical compromise is to use both systems in a *balanced draft* adjusted to maintain atmospheric pressure (or a slight vacuum) in the furnace. Then, expansion cracks, opened doors, etc., will not cause undesirable gas or air flows.

The multivane centrifugal fan is the most common type of draft fan. Backwardly curved blade wheels are usually selected for forced draft service, because of the high speed, suitable for direct motor drive, the self-limiting power demand (a necessary feature when two or more fans are operated in parallel), and high static efficiency. Induced draft fans handle hot chimney gas. Forwardly curved blades which develop a given draft at lower speeds than those with backward curvature are frequently chosen for this service since the speeds and the centrifugal stresses in the wheels will be least.

15–7. Water Pumps. A pump increases the pressure on the water being pumped by an amount sufficient for the service intended. The energy necessary to do this is represented by the equation

$$W = V\Delta P.$$

W = Work, ft. lbs.

V = Volume pumped, cu. ft.

ΔP = Pressure increment, lbs. per sq. ft.

The work is also equal to that which would lift the weight of the water pumped through a height equal to that of a column of water which would create a pressure of ΔP on its base.

Pumps can be classified as follows:

1. Reciprocating pumps.
   a. Direct steam-driven.
   b. Power-driven.
2. Centrifugal pumps.
3. Rotary pumps.
   a. Gear and screw pumps (mostly used for pumping oil).
   b. Propeller pumps.
   c. Positive displacement types (vane and lobe).
   d. Multivane regenerative types.
4. Fluid jet pumps.

Essential data on a pump installation include the head in feet, capacity in gallons per minute, and properties of the liquid such as viscosity, temperature, corrosiveness, grittiness. Secondary data concerning the pump equipment are speed of rotation, power required, and first cost.

A *power pump* is a reciprocating, piston and cylinder type water pump not driven by steam, but by motor, line shaft, belt, or some other mechanical means. The ordinary form is known as a triplex pump. It is three-cylindered, single-acting, having three pistons driven from a single crankshaft. The triplex feature provides fairly uniform discharge, as well as a much steadier load on the motor than would be possible from a single-cylindered pump. The triplex pump has three cylinders in line, and cranks at 120 deg. The crankshaft speed is much lower than the usual speeds of motors and gasoline engines, so that it is best to gear-drive the pump from the source of power. The efficiency of such pumps is rather high, and there is no limit to the pressures which may be carried.

The *centrifugal pump* is selected for all kinds of pumping jobs but is best applied where pressure increments are moderate and flow through the pump is medium to considerable. Very high pressures and small flows are poorly handled by this type. The virtues of the centrifugal pump are compactness, simplicity, quietness, direct drive from motors. Under favorable conditions the hydraulic efficiencies are high. The figure shows that the essential parts of a centrifugal pump are a rotating member called the impeller, and the stationary case surrounding it. The pump action is derived from the conversion of velocity head into pressure head, following the well-known Torricelli formula $v = \sqrt{2gH}$. The liquid to be pumped is let into the pump at the center of the impeller where it is caught in the rotating vanes and caused to take up their angular rotation. This rotation produces a centrifugal force outwardly

directed, and since the water is free to move outward between the vanes of the propeller, it does so, and is thrown from the periphery with considerable velocity. If this velocity were converted to pressure in a haphazard fashion

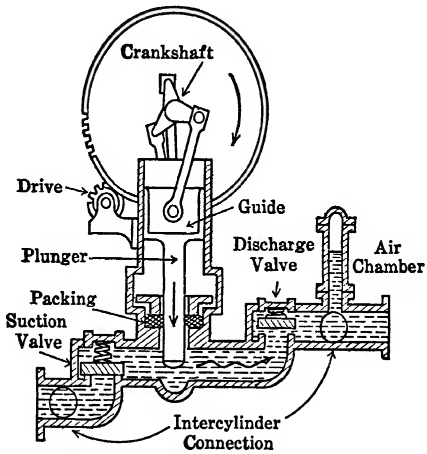
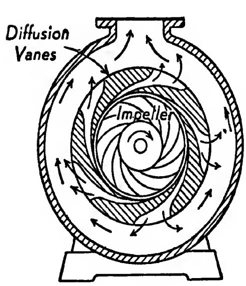


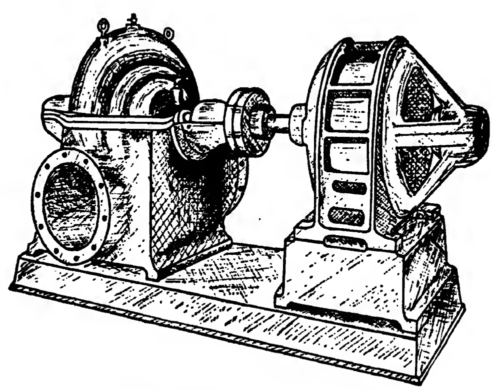
FIG. 15-8. Triplex power pump.

accompanied by eddies, whirlpools, etc., a large portion of the velocity head would be lost in heating effect, and the pressure delivered by the pump would be much less than that which may be achieved through proper control of the water leaving the impeller.

Generally speaking, centrifugal pumps are either of the guide vane or the volute type. Figure 15-9 illustrates a guide-vane type pump, so-called



Section, diffuser type.



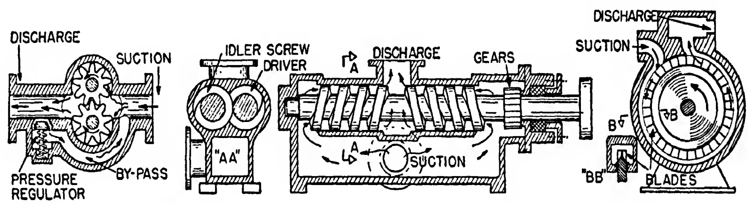
Motor-driven volute type.

FIG. 15-9. Centrifugal pumps.

because conversion of the velocity to pressure head is accomplished by stationary diffusing guide vanes in the expanding passages of which the water loses its velocity and builds up a pressure. It is the more efficient type, espe-

cially on high heads, but is costly. Usually the function of the diffusion vanes is more economically obtained by the use of volute-type casing of proper design. Pumps for heads higher than practical in a single-stage design are built with several impellers in series; that is, although the impellers are mounted on the same shaft, water passes from the first into the second, etc., each impeller producing an additional increment of pressure. Extremely high heads are possible.

The pumps shown in Figure 15–10 are all rotary; *a* and *b* are positive displacement types. Gear and screw pumps are often used with viscous liquids such as oil, the screw type being for larger quantities and higher pressures. The screw pump has two parallel shafts; one power driven, the other idling. They are connected by timing gears and there is no contact between screws.



a. Gear type *b.* Screw, or worm, type *c.* Multi-vane, or turbine, type

FIG. 15–10. Other pump types.

The multi-vane pump employs impact, diffusion, and centrifugal force, but it is not a true centrifugal pump. Liquid enters at the periphery and is carried around to the outlet by the blades, being repeatedly forced out into the free channel only to return for re-engagement with the blades. Considerable churning exists but the pressure rises steadily from inlet to outlet. The efficiency is not remarkable but betters that of the very small centrifugals.

Example 1: A boiler requiring 26,000 lbs. of feed water per hr. is supplied by a motor-driven pump. Find the required capacity of the motor, assuming 60% pump efficiency. Boiler pressure 500 psi. gage, pump suction at approximately atmospheric pressure.

The pumping head due to boiler pressure is so great that it is allowable here to neglect friction in the pipe line from pump to boiler. Consider the pump work as weight multiplied by a lift equal to the pressure head. Since the suction was at 0 psi. gage the pumping head is boiler gage pressure.

$$\text{Height of a column of water equivalent to 500 psi.} = \frac{500 \times 144}{62.5} = 1152 \text{ ft.}$$

$$\text{Work done on the water} = 1152 \times \frac{26{,}000}{60} \text{ ft. lbs. per min.}$$

$$\text{Motor power} = \frac{1152 \times 26{,}000}{60 \times 60\% \times 33{,}000} = 25 \text{ hp.}$$

Example 2: A centrifugal water pump under test discharged 750 gals. per min. Discharge pressure 15 psi. gage, suction 10 in. Hg vacuum. The driving motor re-

ceived $18\frac{1}{2}$ amperes, three-phase alternating current at 440 volts, 85% power factor. Find the overall efficiency of energy conversion.

Energy represented by pumped water can be obtained in this case by the product of pressure and volume.

$$\text{Volume of flow} = 750 \times \frac{8.33}{62.5} = 100.0 \text{ cu. ft. per min.}$$

Net pressure overcome = absolute discharge pressure − absolute suction pressure = $144[(15 + 14.7) - .491(30 - 10)] = 2870$ lbs. per sq. ft.

$$\text{Water power} = \frac{PV}{33{,}000} = \frac{2870 \times 100.0}{33{,}000} = 8.69 \text{ hp.}$$

$$\text{Electrical input} = \sqrt{3}EIf = \sqrt{3} \times 440 \times 18.5 \times .85 = 12{,}000 \text{ watts.}$$

This is the equivalent of $\frac{12{,}000}{746} = 16.1$ hp.

Overall efficiency of energy conversion is therefore $\frac{8.69}{16.1} = 54\%$.

15–8. Piping. Steam and water are carried from place to place in a system of pipes. The piping system, if adequate, ought not to create an appreciable change of state in the fluid. In other words, the pipe conveying a working medium from process to process should be large enough to minimize fluid friction and be covered with enough insulation to render enthalpy changes inappreciable. No matter how excellent the other equipment, the performance of a steam power plant, as a whole, cannot rise above the caliber of its piping system.

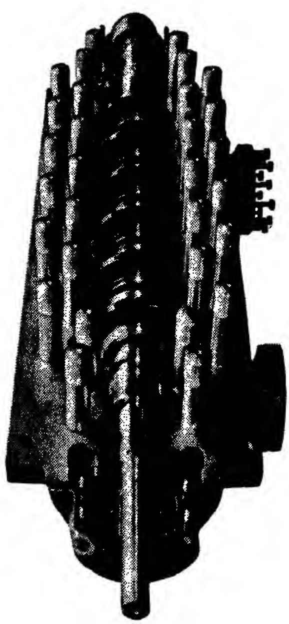


FIG. 15–11. Six-stage centrifugal pump (top half of casing removed). (Courtesy Allis Chalmers Mfg. Co.)

Pipe is usually made of steel, wrought iron, cast iron, copper, or brass. Commercial sizes available, as well as individual physical properties and characteristics, determine the uses and limitations of the various materials. In general, pipe of carbon or alloy steel is used exclusively where high pressure or high temperature, or both, are encountered. The other materials are selected for the lower pressure applications (approximately 300 lbs. per sq. in. and less) where their particular properties may be advantageous. Codes having legal standing and covering recognized good practice frequently govern the engineering and technical details of pipe material selection where public interest or safety is involved.

In order to meet various service conditions economically, steel pipe is manufactured in several wall thickness classifications. Formerly there was a general use of "standard," "extra heavy," etc., to describe wall thickness. The approved system of the American Standards Association introduces a "Schedule Number" to designate pipe weight. The schedule number is 1000 times gage pressure carried, in pounds per square inch divided by working stress, in pounds per square inch (generally between 10% and 15% of the tensile strength of the pipe material). These numbers are made to agree closely with regularly used commercial sizes, the wall thickness of Schedules 40 and 80 being identical with those of the old Standard and Extra Heavy lists up through 8 in. normal size. Pipes are sized by their nominal inside diameter for sizes up to 12 in. When the pipe wall is thickened to provide additional strength, the increment is added to the inside so that, the outside diameter remaining constant, the same size of thread and fittings can be used for all weights. Above 12-in. nominal size the sizes are based on outside diameters.

A typical section of the standard pipe size table is given in the accompanying table for Schedules 40 and 80 up to a size of 6 in.

TABLE 15–1. PIPE STANDARDS (IN INCHES)

Nominal Pipe Size	Outside Diameter	Wall Thickness by Schedule Numbers	
		40	80
$\frac{1}{8}$	0.405	0.068	0.095
$\frac{1}{4}$	0.540	0.088	0.119
$\frac{3}{8}$	0.675	0.091	0.126
$\frac{1}{2}$	0.840	0.109	0.147
$\frac{3}{4}$	1.050	0.113	0.154
1	1.315	0.133	0.179
$1\frac{1}{4}$	1.660	0.140	0.191
$1\frac{1}{2}$	1.900	0.145	0.200
2	2.375	0.154	0.218
$2\frac{1}{2}$	2.875	0.203	0.276
3	3.5	0.216	0.300
$3\frac{1}{2}$	4.00	0.226	0.318
4	4.5	0.237	0.337
5	5.563	0.258	0.375
6	6.625	0.280	0.432

A piping system, in fulfilling its function of providing a flow path for liquids or vapors, is rarely a straight run of pipe between two points. Flows are joined, parted, started, stopped, and regulated in the piping system.

Only occasionally is it possible to take a "crow flight" path between end connections; the common run of pipe must follow configurations of equipment, walls, floors, beams, etc. Fittings and valves, properly incorporated in the pipe system, enable it to meet these varied service conditions.

Adjacent sections of pipe are connected together in various ways, but all connections can be grouped together under these four headings.

1. Packed and caulked joints, such as leaded bell and spigot, or plain-end coupling. Mainly used for low pressures, soil pipe, drainage pipe.
2. Screwed joints, such as couplings and unions. Generally used for sizes less than 4 in.
3. Flanged joints, with companion flanges either loose or screwed, shrunk, riveted, or welded to the pipe. Flanged fittings are generally called for in the larger sizes of pipes; and for high-pressure high-temperature work they have entirely superseded screwed connections. The flanges are bolted together with a sealing gasket between them.
4. Welded joints. Welds made by the fusion process, using gas or electric welders.

Most flows of steam through piping may be considered incompressible. If an incompressible fluid flows steadily in a pipe line past two stations where the cross-sectional areas are A_1 and A_2, then the mean fluid velocities v_1 and v_2 are related thus:

$$A_1 v_1 = A_2 v_2.$$

Piping carrying fluid will have a frictional surface with the fluid. The extent to which friction is developed at this surface is a matter governing either the size of pipe which would be used, or the reduction by friction of pressure or velocity head in the pipe. The size of a pipe is not determined alone by the weight or volume of the fluid being transported. For instance, there is no one pipe size that must be selected to carry 500 cu. ft. of steam per min. At 5000 ft. per min. velocity a pipe of 0.1 sq. ft. cross-sectional area is required; at 10,000 ft. per min. it is 0.05 sq. ft. As soon as the velocity and the volume are known, the pipe size is determined by this relation.

$$\text{Flow volume} = Av.$$

TABLE 15–2. AVERAGE PRACTICE IN FLOW VELOCITIES (FEET PER MINUTE)

Water	300–600
High-pressure saturated steam	5,000–10,000
High-pressure superheated steam	10,000–15,000
Atmospheric exhaust steam	8,000–12,000
Low-pressure exhaust steam	20,000–24,000

Power plant piping is comparatively short in length, but usually liberally endowed with elbows, valves, and other impediments to flow in which the friction loss greatly exceeds that of the connecting straight lengths of pipe. As the friction loss in these fittings can be only roughly approximated, there is but little advantage in detailed hydraulic calculation of fluid friction in straight pipes. The friction loss of installed piping is measurable with pressure gages or manometers, while to estimate proper pipe sizes for new installations, flow velocities known by experience to be satisfactory can be assumed. Velocity and quantity flowing are sufficient to fix pipe size. Table 15–2 is an experience table of flow velocities.

15–9. Steam Trap. A device which automatically operates to allow the discharge of water from a certain region, and prevent the escape of steam, is known as a trap. The trap is used when condensate is to be drained from a vessel occupied by condensing steam, without the loss of steam. It traps the steam in the vessel and passes the water of condensation. Traps are used with steam heaters and cookers of all types, wherein a surface is interposed between the steam and that which is heated, also to drain the condensation from steam lines, and for many similar services. A great deal of heat which is otherwise wasted by a partially opened drain valve will be saved by a trap if its discharge is connected to some point where the heat in the hot condensate can be used. The principal methods of operating a steam trap are expansion, float, tilting under the influence of accumulated condensate, and sinking bucket.

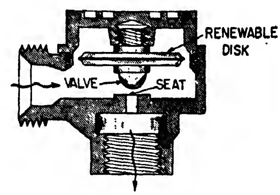


Fig. 15–12. Radiator trap. Disk is a hollow thermostatic element which will swell in the presence of steam and force the valve against the seat. Condensate will cool it and the valve will open.

In an expansion trap, advantage is taken of the fact that condensate is usually a little cooler than the steam, and a metal expansion element is placed where it may be covered either with steam or condensate, depending on the amount of condensate in the trap. When it is covered with condensate it is cooler, and shrinks, opening a valve to pass the condensate from the trap. As the condensate is discharged, the expansion element is uncovered, and the hot steam causes it to expand and close the discharge valve. Such traps are frequently used on steam radiators. In a float-type trap, the float is attached to the end of a pivoted lever, from the other end of which a link bar extends to open or close a discharge valve, depending on the position of the float. These two types tend to produce a continuous discharge through a valve which is cracked open just sufficiently to maintain a condition of equilibrium in the water level of the trap. The tilting- and bucket-type traps are intermittent in operation, which is much better from the standpoint of wear on the discharge valve. In the tilting trap, water collects in a pivoted chamber

until the weight of it overcomes the counterbalance and the whole chamber tilts until a valve is opened and the discharge of water through it relieves the trap so that the counterbalance can return it to the closed position. A pivoted bucket trap is shown in the accompanying diagram. As the condensate is collected in the trap, it fills the bucket, which otherwise would tend to float on the surface of the water in the bucket chamber. When the bucket is filled it sinks, thereby opening a discharge valve. The pressure of the steam forces the water out through the discharge valve, and as soon as the bucket is empty, it again floats on the water in the chamber, closing the discharge valve.

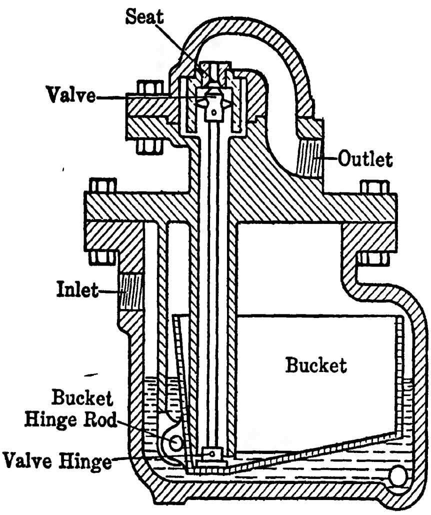


FIG. 15–13. Intermittent discharge trap (bucket rising, following discharge).

15–10. Generation of Electrical Energy. Electrical energy is so much more convenient to distribute, so readily and efficiently reconverted to mechanical work, and so useful for other services, like lighting, heating, communication, etc., that extensive amounts of it are generated from the mechanical work of prime movers. The greatest use of steam turbines is to furnish the mechanical work for transformation to electrical energy. The turbine is usually direct-coupled to the electric generator, as illustrated in Figure 14–16.

The alternating-current circuit is that type which carries electrical current which rapidly reverses in direction of flow. Most electrical circuits are now for alternating current, although public service began with direct current systems. Alternating-current generation has received great impetus because of the simple way in which it can be changed in voltage. This is of great advantage in transmitting electrical energy over long distances, since the cost of transmission lines decreases when the current carried is at high voltages. The advantages of the induction motor are also a point in favor of alternating current, but alternating current is not without defect, since inductance and capacitance, factors unknown to direct current circuits, are present. These account for the less-than-unitary power factors, the effect of which is apparent on page 93. The frequency of an alternating-current circuit is the number of complete alternations per second. Sixty cycles per second constitutes the standard frequency in the United States.

An electromotive force is generated in a conductor when some relative motion causes it to "cut" the lines of force between the poles of a magnet.

The elementary principle of a simple two-pole, single-phase alternating-current generator is shown. When the magnet revolves it will carry with it lines of force that will cut the conductor, which is a wire loop embedded in the stationary portion called the armature, and will generate an alternating current.

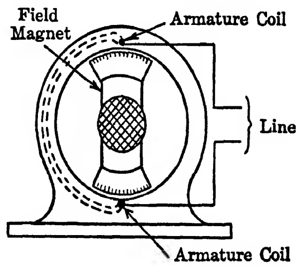


Fig. 15–14. Elementary alternating-current generator.

This elementary principle must be expanded in several directions, if a practical generator of alternating current is to be had. First, the rotating part, or rotor, must have magnetic strength in excess of that which could be obtained from a simple permanent magnet. In other words, the poles must be formed by electromagnets whose energy, in the form of direct current, must be carried to the rotor through slip ring connections. The rotor is called the field, and the current it uses is called the field current. Occasionally as few as two poles are used, but since this requires a very high rotative speed, four or more poles are much more frequently employed. Sixty-cycle current will be generated at 1800 rpm. by four poles; at 3600 rpm. by two poles.

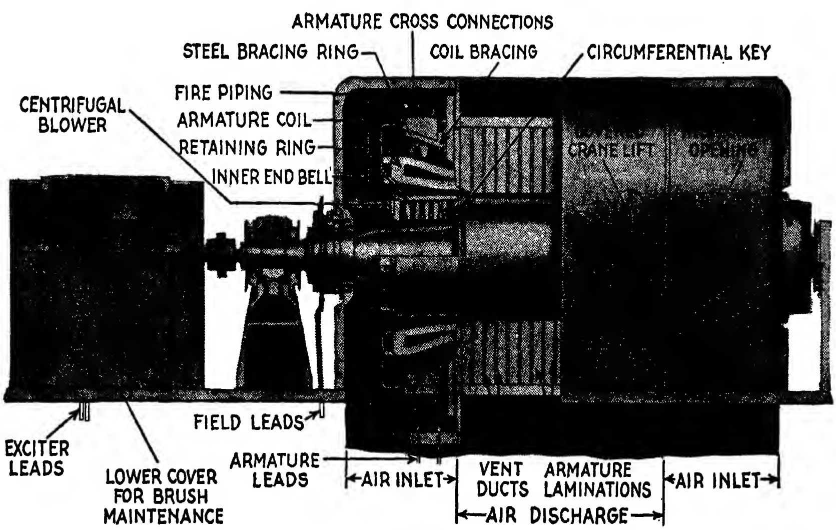


Fig. 15–15. Half-section view, showing typical construction of generator and direct-connected exciter. (Courtesy Westinghouse Electric and Mfg. Co.)

Basically, the electrical generator is a device for converting mechanical into electrical energy. While it is able to do this with a high degree of efficiency, it does suffer the following losses:

1. Friction and windage from bearings and fan action of the rotor.

2. Core loss, which is the result of magnetic eddy currents and hysteresis in the iron core.
3. Resistance heating loss in the armature and field conductors.
4. Resistance loss of the field rheostat.

Practically all generator losses appear as heat in and about the windings, and to maintain these at a safe working temperature, a cooling medium must be employed. Air has been the medium generally used. The rotor may or

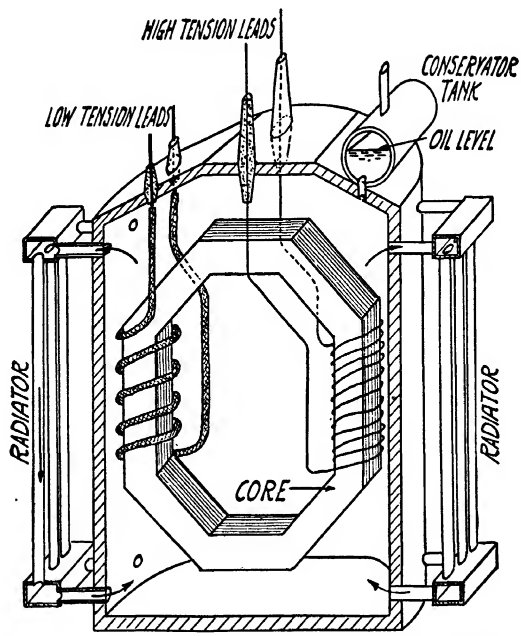


Fig. 15–16. Schematic sectional view of a single-phase, self-cooled transformer.

may not be able to produce its own fan action, depending on the size, speed, and construction. Ventilating air frequently has to be brought through a duct and discharged through the interior of the generator. To supply the current for the field, a source of direct current at 110 or 220 volts is necessary. This is delivered from a small direct-current generator called the *exciter*. The exciter may be driven from an extension of the alternator shaft, or it may be driven by some independent means, such as motor or engine.

15–11. Electric Power Transmission. If mechanical power could be generated for the same cost at any point in the country, there would be little need for electric power transmission. But since the larger the generating unit, the lower the unit cost of production, there has developed the method, so extensively used at present, of generating energy in large central stations. From this has arisen the need for electric power transmission to carry the energy so produced to the users.

Energy may be transmitted electrically in overhead wires or underground cables. It cannot be transmitted without some losses, the principal one being a resistance loss, which depends upon the current flow and the size of the wire. In transmitting a given amount of energy, this loss may be reduced by increasing the voltage, since any voltage increase will allow a decrease of current. Or, from another viewpoint, for a given amount of energy, and a given permissible loss, higher voltages will permit the use of smaller wires. The foregoing should serve to show that, where practicable, electric power should be transmitted at high voltages.

These economic transmission voltages are higher than those under which power generating and utilizing apparatus can be operated, and a voltage transformation is essential if high transmission line voltages are to be used. By far the simplest and most effective means of accomplishing this is the electrical "transformer." The transformer may be used only with alternating current, and this goes far in explaining the prevalence of the alternating-current method of power transmission, because in many other respects high voltage direct current is a superior means of transmission.

Without doubt, adoption of alternating current in favor of direct current by the growing electric industry was due to the ease and efficiency with which electric energy could be transferred from low to high potential, and vice versa. The power *transformer* is used to increase generator voltage to economical transmission voltage or to

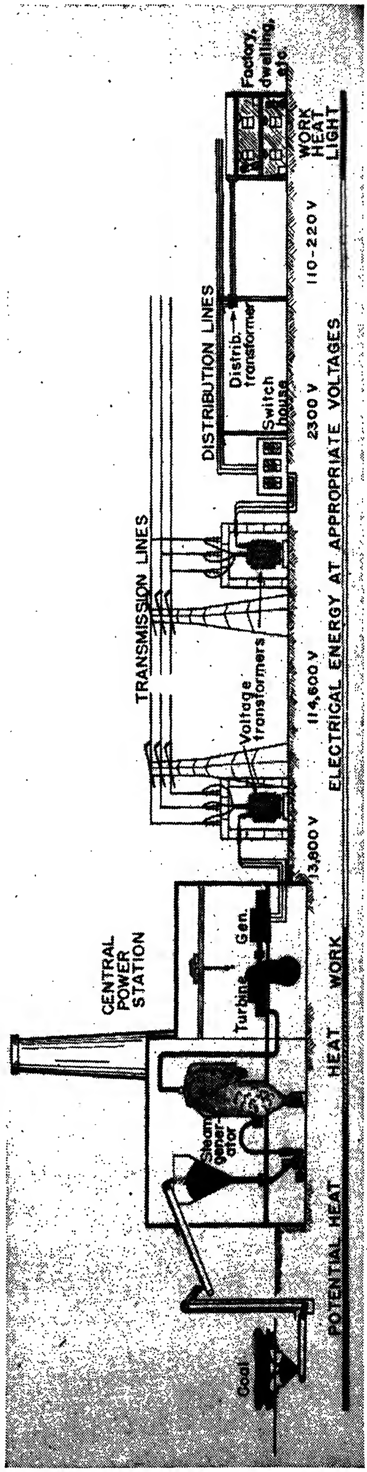


Fig. 15–17. Energy flow in an electric power system.

decrease generator or line voltage for power service. It can be defined as a device for transferring electric energy from a circuit at one voltage to one at another. There are no moving parts, nor is there any electrical connection of the two circuits. The energy is transferred through magnetic linkage. Regardless of the voltage, the energy supply circuit is termed the primary, and the energy receiving circuit the secondary.

The ordinary single-phase transformer consists of the magnetic circuit, the windings, the leads, the insulating bushings, the insulating oil and its cooling system, and the tank to contain the oil and provide support for the other components. In principle, the three-phase transformer is similar. The losses which occur in a transformer are core losses and resistance losses. The core losses are nearly constant and are present as long as the transformer is connected to the supply circuit.

With transformers, transmission line circuits, switches, and protection—such as lightning arresters, circuit breakers, fuses, etc.—the energy which once resided dormantly in fuels can be transmitted miles from the region of combustion.

15–12. Electric Motors. An electric motor is a machine which, receiving energy in the form of electricity, converts it into mechanical form. Common motor types can be classified as:

1. Direct current.
2. Three-phase alternating current.
3. Single-phase alternating current.

Electric motors are built in a range varying from outputs of $\frac{1}{100}$ hp. up to well over 1000 hp. A 50-hp. motor is considered a large one, and most electric motors now in use range between $\frac{1}{4}$ and 10 hp. Standard motor sizes above the small fractional sizes are $\frac{1}{4}$, $\frac{1}{3}$, $\frac{1}{2}$, $\frac{3}{4}$, 1, $1\frac{1}{2}$, 2, 3, 5, $7\frac{1}{2}$, 10, 15, 20, 25, 30, 40, and 50 hp. Standard 60-cycle synchronous speeds are 3600, 1800, 1200, 900, 720, 600, 514, and 450 rpm. Full load induction motor speeds are 2% to 5% less than these. The efficiency of the electric motor ranges from 75% to 95%. It is higher in large motors than in small. Induction motors are more efficient the higher the rated speed, but direct-current motor efficiency is little affected by speed. Efficiency, however, is often secondary to reliability. Direct-current motors are much less frequently employed than alternating current, because of the preponderance of alternating current over direct-current systems. However, speed control and starting torque are so excellent with direct current that it is frequently used where these characteristics are important.

The losses sustained by a motor in converting electrical to mechanical power arise chiefly through the electrical and magnetic characteristics. The mechanical simplicity of a motor enables it to be designed with almost negligi-

ble friction. The losses are, then, the resistance losses occasioned by current flowing through the conductors of the armature, the field, or the controller, and the core losses of hysteresis and eddy currents. The cores of all motors must be built up of laminations insulated from one another by lacquer or enamel, otherwise this core loss becomes excessive. The motor is so compact that in large sizes, although the efficiencies are high, the heat liberation per unit volume becomes sufficient to need the positive ventilation secured by fans and impellers. In the small open frame motor the windage of the motor itself is generally sufficient for cooling. Larger sizes are cooled by air forced through the windings by impellers mounted on the motor shaft, or by external fans.

Important individual characteristics of electric motors include starting torque, normal speed, speed regulation, reversability, efficiency, and cost. Motors are selected upon the basis of the voltage available, the peak load, the necessary reliability, the desired speed range, and the load factor.

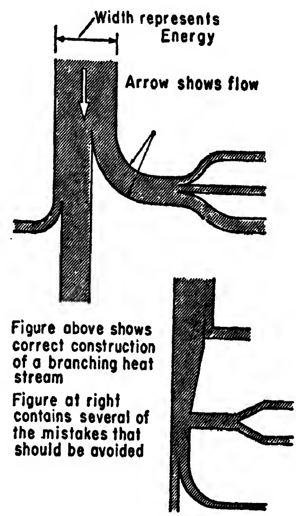


FIG. 15-18. Principles of heat stream construction.

15-13. Heat Balance. A heat balance is a method of accounting for all energy in a process during which heat is transferred or transformed. Examples of cases where heat balances might be undertaken are: (1) determining the nature and the magnitude of the various losses which occur when coal is burned in a steam boiler furnace; (2) accounting for all energy involved in the operation of a prime mover such as a Diesel engine or a steam turbine; (3) determining the distribution of heat in a static heating device such as a water heater supplied with steam.

Heat balance work is based upon the first law of thermodynamics. The significance of this law applied to the heat balance is that the total energy may be accounted for by straight addition, hence striking a heat balance resembles bookkeeping, with energy supplied on the credit side of the ledger, and energy discharged on the debit side. One way of showing a heat balance is a tabular form, another shows the energy as a stream, properly branched and subdivided to indicate the distribution. Obviously this balance may often involve other energy forms than heat, but *heat balance* is used advisedly, as that form predominates in cases where the term is applicable.

A heat stream is often the best way to present a heat balance, as it enables the reader to grasp at once the relative magnitude of energy involved in the different parts of heat-power equipment. To construct this diagram one must have at hand a calculated breakdown of all energy quantities, being sure to

test the calculations at all points possible for obedience to the famous *first law*. In other words, we must be careful to account for *all* energy. The widths of the heat stream represent energy graphically so the scale could be any unit of energy. A common basis, where combustion supplies the heat, is to consider the heating value of the fuel as 100% and have the widths of the energy streams as percentages of that heat.

Since stream width is energy, any tapering of the stream is in violation of the first law. Streams may be split off or added together, but they may not be tapered. So, where streams are to be curved, it is important that the same

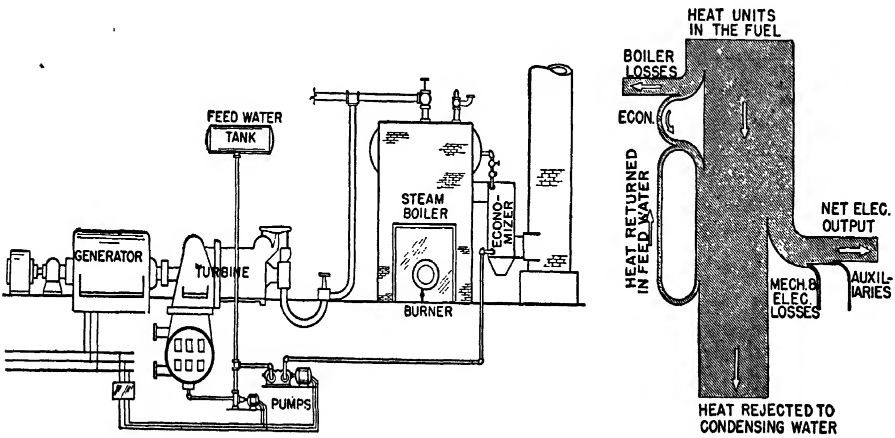


FIG. 15–19. Simple condensing steam power plant.

center be used for both inner and outer arcs. Never use square corners for heat streams, for the width perpendicular to the center line would not be constant.

The author knows of no better way to conclude this survey of applied energy than to show the flow diagram of a steam power plant, and with it a heat stream that was calculated from test data taken at the plant. The data and calculations were, necessarily, extensive, but note how clearly the relative energy distribution becomes when the results are finally transcribed to a *heat stream.*

PROBLEMS

1. The temperature of flue gas leaving a boiler is 850° F. Air-fuel ratio is 12:1. It is desired to reduce the final gas temperature to 500° F by the use of an air preheater which will take in atmospheric air at 50° F and give it a preheat. If all the air used for combustion is preheated, what temperature will it have at the preheater outlet? Specific heats of air and gas are 0.24 and 0.26 B.t.u. per lb. per deg., respectively.

2. Only the secondary air going to a boiler furnace is preheated. Preheater is to raise the air temperature by 225° F. Primary air is 15%. Boiler gas outlet tempera-

ture, 900° F. Air-fuel ratio 11:1. Specific heats of air and gas, 0.24 and 0.25 B.t.u. per lb. per deg., respectively. Estimate the chimney temperature.

3. The air-fuel ratio in a boiler furnace is 14:1, rate of evaporation 10 lbs. of steam per lb. fuel. The flue gas passing the economizer drops 250° in temperature. Assuming that the gas's specific heat is 0.25 B.t.u. per lb. per deg., estimate the temperature rise of the water in the economizer.

4. Which figures of Chapter 13 show an economizer present? Which show an air preheater?

5. The water cylinder of a direct-acting pump is 4 in. in diameter. Both water and steam cylinders are connected to a boiler drum where the pressure is 200 psi. gage. If the steam cylinder diameter is 1.6 times that of the water cylinder, how much thrust is thus made available to operate the pump? List the physical resistances that consume this thrust.

6. A direct-acting steam pump uses dry and saturated steam at 100 psi. abs.; exhaust to atmosphere. It pumps 30 gals. of water per min. against a head of 250 ft., using meanwhile, 6.2 lbs. steam per min. What was its "duty"? Note: B.t.u. "chargeable" per pound of steam is enthalpy at throttle less heat of the liquid at exhaust pressure. The h_f is credited because presumably it might be recovered by condensation of the exhaust.

7. The exhaust of a steam engine is used to heat boiler feed water in an open heater. The 50-hp. engine operates with a steam rate of 56 lbs. per hp. hr., using steam at 100 psi. gage dry and saturated. Water comes to the heater at 60° F.

a. Use the steam rate to calculate the B.t.u. removed from each pound of steam and converted into work by the engine.
b. How many B.t.u. per minute are there in the engine exhaust steam? Neglect energy diverted to friction and radiation.
c. How much feed water can be discharged from the heater, lbs. per min., at a temperature of 200° F?
d. Sketch the heater and record all data and results on the sketch.

8. A surface-type heater is supplied with dry and saturated steam at 25 psi. gage. Water enters it at 150° F and is heated to within 10° F of the steam temperature. Condensate drains out 10° cooler than the steam. How much water can be heated with 2500 lbs. of steam? Diagram the heater and record all quantities thereon.

9. A central station using Indiana bituminous coal burns it in pulverized form, the average rate being 11,500 lbs. per hr. 20% of the ash is unprecipitated fly ash. How many pounds of ash are discharged from the chimney per year? If this settled evenly on 4 sq. miles of surrounding land, what is the deposit per square foot?

10. Given chimney height 150 ft., average chimney temperature 500° F. Molecular weight of flue gas, 30.

a. Find the draft in pounds per square foot on a 30° day, and on a 90° day.
b. Express the results of part (a) in "inches of water."

11. The molecular weights of air and flue gas are 28.9 and 30, respectively, and the temperatures likewise 65° F and 600° F. How high a chimney is required to produce a draft of 1.55 in. water? (A column of water 12 in. high produces a load of 62.4 lbs. on each sq. ft. of base area.)

12. In a breeching of expanding cross-section, the end areas are 48 in. × 48 in. at entrance, 100 in. × 60 in. at exit. Flow 24 lbs. per sec. Temperature at entrance 820°, at exit 800°. Molecular weight of flue gas, 30. Omit effect of surface friction.

What is the pressure increase (inches of water) caused by this diffusion? Would it aid or load the draft system?

13. A forced draft fan is drawing air from the boiler room and sending it to a preheater at the rate of 25,000 lbs. per hr. Boiler room temperature 80° F. Velocity in discharge duct 35 ft. per sec., static pressure 4 in. water plenum. What pressure boost is produced by the fan? Include velocity head.

14. A centrifugal pump is used to withdraw condensate from a condenser and deliver it to an open overhead storage tank. Difference in water levels of storage tank and condenser hotwell, 30 ft. Condenser vacuum, 28 in. Hg. Allow 1.5-ft. head friction loss in piping and determine operating head on the pump in (a) ft. of water, (b) lbs. per sq. in. Neglect velocity head.

15. A boiler feed pump for 600 psi. gage pressure is supplied with water at 20 psi. abs. Rate of flow 37 gals. per min. Pump efficiency 86%. Estimate the required driving power.

16. A water flow of 180 gals. per min. is carried through a 3-in. schedule 80 pipe. What is the exact velocity of flow, ft. per sec.?

17. What size pipe (schedule 40) should be used to bring the steam to the heater of Problem 8 with a velocity not exceeding 10,000 ft. per min.?

18. The steam header of a power plant is measured and found to have an outside circumference, over the bare metal, of $14\frac{1}{4}$ in. Steam condition is 175 psi., 450° F. Assume that it is "extra heavy" grade of pipe. What weight of steam can be transmitted per hour at a velocity not to exceed the average cited in Table 15–2?

19. Study actual examples of the following types of pipe joints and diagram them by means of a sectional view.

a. Screwed union (not a "coupling").
b. Flanged joint.

20. The balanced three-phase output of an alternating current generator is described as follows: 13,200 volts, 60 amperes, 80% power factor. Losses in generator core and windings are 3200 B.t.u. per min. Exciter is direct-connected also. What is the turbine output (horsepower)? Why is it that exciter input does not have to be considered in addition to the above data? What generator loss is neglected in this analysis?

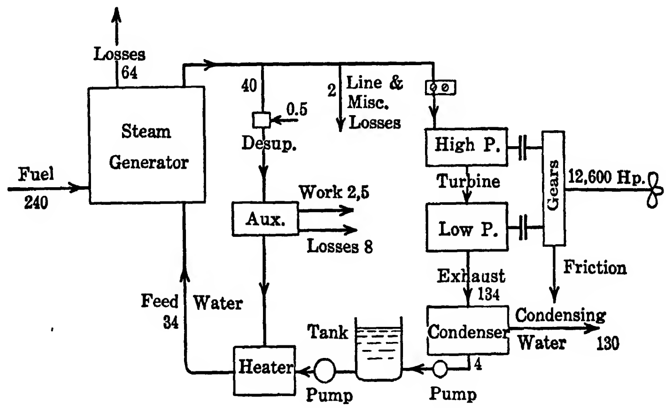
21. A unit like that shown in Figure 14–16 is driven by a turbine developing 5000 shaft hp. Eight thousand cu. ft. per min. ventilating air is blown through the generator, with a temperature rise of 70° F. Find the electrical output, kilowatts. What losses does this analysis neglect?

22. Calculate the full load line current of a 20-hp., 1200-rpm., 220-volt, single-phase motor. Efficiency 88%, power factor 80%.

23. What is the efficiency of a direct-current motor operated at 110 volts? When delivering $\frac{3}{4}$ hp. the line current is 6.4 amperes.

24. Construct a scaled heat stream for the energy flow of a Diesel engine. Let the incoming heat in the fuel be represented by a stream $2\frac{1}{2}$ in. wide. Diagram should occupy nearly a full page. Brake horsepower 500. Fuel rate 0.42 lbs. per hp. hr. H.H.V. of fuel 18,900 B.t.u. per lb., L.H.V. 18,100 B.t.u. per lb. Losses (based on L.H.V.) are cooling 30%, exhaust (sensible heat) 25%, incomplete combustion 5%, radiation 1%. Friction, auxiliaries, and miscellaneous account for the balance. Color or shade the heat stream; also label fully.

25. The accompanying figure is a simplification of the marine steam power plant shown in Figure 12–14. The numbers represent energy flow in units of millions of B.t.u. per hour. Assume that these have been determined from tests by the chief



engineer, and that you are directed to construct a "heat stream" of energy flow, using a scale of 1 in. = 50% of the incoming heat in the fuel. Drawing should occupy a full page, and the stream be appropriately colored or shaded, and be fully labelled.

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Class No. 62L101 Book No. 83E
Vol.

Author... Morse

Title Applied energy

Acc. No. 36077

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